FM Global
Property Loss Prevention Data Sheets 13-3
January 2001

STEAM TURBINES

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1.0 SCOPE

This data sheet consolidates the steam turbine information formerly contained in Data Sheets 13-3, 13-9 and 13-11. It provides loss prevention information for electric power generation and mechanical drive applications. Driven machines are dealt with in separate documents, some of which are noted in Section 4.0, References.

This document contains loss prevention recommendations applicable to physical protection, testing and dismantle inspection and substantiation for the recommendations. In some cases, the technical material necessary for carrying out the recommendations is also provided. Unless otherwise specified, the loss prevention recommendations in this data sheet apply to all steam turbines, regardless of size.

For use in this data sheet, a steam turbine is defined as the base turbine (stationary and rotating elements) up to the driven object coupling flange, and includes valves, piping, oil systems and monitoring and control systems as noted below.

- Valves
  Main steam (stop and control)
  Reheat steam combined stop and intercept
  Extraction stop and non-return
  Feedwater heater (FWH) level control
  Steam line and lead drain
- Piping (up to the isolation valves and including all hangers and connectors)
- Monitoring and Control Systems (I&C Systems)
  Governor Speed Control
  Overspeed
  Vibration
  Temperature
  Steam Pressure
  Water Quality/Steam Purity
  FWH Level Control
  Condenser Vacuum
- Oil Systems
  Lubricating
  Hydraulic (control oil)

In the interest of clarity and consistency, the steam turbines covered in this standard are described as follows:

- *Generation company units* — Steam turbines coupled to electric generators at facilities whose principle function is to produce electric power for sale to private and commercial entities including transmission and distribution companies.
- *Industrial units* – Steam turbines coupled to electric generators whose principal function is to produce electric power and thermal by-products to an adjacent host (or other entities). Excess power may be sold to the local grid.
- *Mechanical drive units* – Steam turbines driving equipment other than generators.

This standard also contains extensive discussion of generation company (formerly utility) unit steam turbines in cyclic or peaking service. Based on an FM Global study of industry control practices, it was found that general guidelines could not be provided for inspection of turbines in this type of service. Therefore, it is felt that FM Global can only recommend that control be based on cyclic life analysis of a given rotor and that the percent life used during each cycle of operation should be established. This is usually the responsibility of the manufacturer, but may be accomplished by any qualified person(s).

FM Global recognizes that several customers are applying condition–based, reliability–based and risk–based methodologies to their facilities or limited systems or equipment within them to extend inspection intervals and reduce overall O&M costs. Technical judgment is required when assessing alternatives to the practices described herein. FM Global will consider alternative approaches if the customer can demonstrate the risk of loss will not increase.

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1.1 Changes

Recommendation 2.2.2, Overspeed Trip Tests, has been revised to incorporate the latest overspeed trip technology.

All information applicable to electric generators can be found in Data Sheet 5-12, *Electric AC Generators*.

Pertinent manufacturers' technical information is provided in Section 2.3, Alert.

1.2 Superseded Information

This document supersedes Data Sheets 13-3, 13-9 and 13-11 and Inspection Guides 2 and 5.

2.0 LOSS PREVENTION RECOMMENDATIONS

Major maintenance on large turbo-sets is very expensive. Total cost includes both direct and indirect costs associated with labor and materials, as well the indirect cost of lost generation or process capacity while a unit is out of service. Extension of the time between major maintenance outages can reduce operating and maintenance costs, however, failure to carry out maintenance can result in failures with serious consequences that may lead to costs that are substantially greater than planned maintenance costs.

Minimum loss prevention measures from FM Global's point of view consist of:

- Vibration monitoring
- Performance monitoring
- Chemical control (water and steam purity)
- Material inspection
- Lube oil system monitoring

These are addressed in the following section.

2.1 Equipment and Processes

2.1.1 Protective Devices

Table 1 lists protective devices, alarms and trips that are recommended for steam turbines. The functioning of some of these is discussed in Section 3.2, Hazards to steam turbines. Use the turbine manufacturer's recommended alarm and trip settings.

A category entitled ''Advisory Practices'' is included in the table. Although advisory in nature, these practices can play a substantial role in loss prevention. If conditions at a given installation warrant, these should be applied as recommendations.

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Table 1. Protective Devices, Alarms and Trips

A. Recommendations			
Protective Device	Actuated Device	Alarm	Trip*
1. Emergency overspeed trip	Turbine steam admission emergency stop valves		x
2. Thrust-bearing axial position monitoring by means of proximity probes, hydraulic sensors, micrometer measurement, or temperature detectors in thrust bearing. Advisory for mech. Drive turbines (5≤25 MW)	Emergency shutoff valve solenoid	x	x
3a. Low oil pressure sensor or emergency lube oil pumps.	1. Annunciator	x	
	2. Emergency shutoff valve solenoid		x
3b. Low oil level sensor in tank (gravity systems)		x	
4. Condenser low vacuum	Emergency shutoff valve	x	x
5. Automatic drains at low points and before and after valves in inlet steam lines			
6a. Vibration instrumentation installed on all bearings (electric power generating units B-T-G sets)	Alarm in constantly attended location Senior Operations Notification Procedure.	x May have 2nd high level alarm	
6b. Vibration monitoring by hand-held or installed instrumentation (<20MW)	Senior Operations Notification Procedure	x	
7. Pressure relief valve between turbine and any discharge block valve (mech. drive turbines)			
8. Emergency lube oil pump (backup for auxiliary lube oil pump).			
9. High-level switches on all extraction feed water heaters	High level drain valve to the condenser or other receiver, trip NRV solenoid	x	
10. High-high-level switches on all extraction feedwater heaters	Electrically operated block valve in extraction line, power operated drain valve on the turbine side of the NRV	x	
	Or		
	Automatic shutoff valves on all sources of water to feedwater heater	x	
11. High water-level switch on boiler drum*(B-T-G units)		x* Alarm recommended in Data Sheet 12-17	
B. Advisory Practices			
Protective Device	Actuated Device	Alarm	Trip
1. High oil temperature sensor		x (180-190°F 80-85°C)	
2a. Vibration instrumentation on all bearings (≥20 MW)	Emergency shutoff valve solenoid	x	x
2b. Vibration instrumentation on all bearings (>10 MW to <20 MW)**	Alarm & senior operations notification procedure	x	
3. Thermocouples in main steam line to sense water induction	Emergency shutoff valve solenoid	x ≤20 MW	x >20 MW

* Sequential trip (turbine, followed by generator breaker when stop valve limit switches show valve is closed).
** Also applies to any units considered to be critical to plant operations.

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2.2 Operation and Maintenance

2.2.1 Turbine Shutoff Valves

The main steam shutoff valves immediately preceding the control valves are variously called the stop valves, the emergency stop valves and the turbine shutoff valves. For purposes of this data sheet, emergency stop valve will be used for consistency.

Many large units include a valve between the main steam outlet and the emergency stop valve(s). This is the main steam block valve. It is used to permit maintenance on the turbine and accessories when the boiler is pressurized. It can also be as the boiler code termination point.

Dismantle, inspect and refurbish the emergency stop (steam shutoff), intercept, and extraction-line nonreturn valves at least every five years or more often if conditions warrant because of abnormal wear and deterioration.

Based on the particular type of valve, the scope of these inspections includes:

- valve disassembly
- cleaning of valve parts
- measurement of stem or shaft to bushing clearance
- nondestructive examination of stems (or shafts), disks, arms, pilot valves and seats
- total indicator runout measurement of stems (or shafts) and valve seats to check for warpage
- measurement of clearance at valve-guide seal rings
- valve seat contact check
- replacement of all fasteners, washers, seals and similar hardware
- correction of all out-of-blueprint measurements
- inspection of screens
- nondestructive examination of bolts and studs
- correction of all discrepancies

Exercise the emergency stop valve and/or the trip mechanism at least weekly while the turbine is under load. Exercise the emergency stop valve prior to each overspeed test (Section 2.2.2). For those valves not equipped with exercisers, design permitting, check freedom of valve to operate by manually stroking it a percentage of its travel.

Exercise the extraction line nonreturn valves weekly, by hand, or using the air-operated assist mechanism, to ensure freedom of operation of the valve-disk shaft Perform an actual leakage test in accordance with valve manufacturer specifications, or dismantle and inspect at least every five years; annually where exercisers are not installed.

2.2.2 Overspeed Trip Tests

The first line of defense in preventing losses due to overspeed is the turbine governor system that trips the control valves closed on detection of approximately 103% of rated speed. The control valves will re-open if the speed returns to normal. However, if the speed continues to increase, the emergency overspeed trip system will be activated when a preset speed level is reached.

A formal documented overspeed trip testing procedure that incorporates specific turbine manufacturer's guidelines and requires direct owner engineering and/or supervisory oversight is needed for all steam turbines. The procedure shall include the recording of completed overspeed trip test results and include a section for comments describing any aborted tests or other test difficulties experienced. Operating personnel should have documented proficiency in the procedure and control logic.

Test turbine overspeed trip systems as described in sub-sections (a) through (e) below to confirm operability.

Redundancy is an essential factor when considering emergency overspeed trip system reliability. Electronic systems having two-out-of-three voting logic are preferred. These systems normally alarm in the event

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of the loss of one channel and trip on the loss of a second channel. Such systems are considered to be fail-safe because loss of power actuates the emergency trip solenoid open to dump oil to drain, which, in turn, trips the emergency stop valves.

Subsystems of the electronic trip system can be periodically tested on- or off-line using procedures and frequency approved by the supplier. To verify operability following an outage, test the complete system prior to returning the turbine to normal service. This is especially important where any portion of the trip system has been worked on.

It is ***not acceptable*** to use any trip systems that could be subjected to a common mode failure, such as a single trip system having two independent power sources or two trip systems having a common power supply.

See Section 2.3.2, Inspection Interval Support for Overspeed Protection.

(a) Mechanical Trip Systems. Conduct a controlled overspeed trip test at least annually. Recommended practice is to perform two tests on each system, with the maximum speeds within the tolerances specified by the manufacturer. If one test is out of tolerance, the overspeed test is considered to have failed.

Troubleshoot the overspeed trip and emergency stop valve system to determine and correct the cause of the failed test. Following this, conduct two additional tests. Document all test results, including failed tests.

Mechanical overspeed sensing devices are prone to wear, binding and corrosion, all of which may affect their reliability. The mechanical overspeed sensing devices presently used have been in use for many years with only minimal design changes. These devices have always had inherent problems requiring their frequent testing. It is strongly suggested that, when feasible, these systems be replaced with electronic trip systems.

(b) Electronic Trip Systems. Electronic trip systems may be tested using their built-in simulation systems. Conduct a full overspeed trip test where no simulation system is provided.

(c) Alternative Trip Systems. Alternative turbine trip systems include those equipped with electronic overspeed systems having two or more independent trip devices, each having independent power supplies [ac/dc], or a combination of an electronic system and a mechanical system. Test mechanical systems annually. (See paragraph a).

Some vendor systems provide for off-line testing of the complete system with the turbine at “0” speed by substituting frequency generators in place of the speed probes. System suppliers and turbine suppliers do not believe actual overspeed testing is required for those machines equipped with off-line testing. Off-line testing has the benefit of reducing life cycle stress. *(Applicable to paragraphs b and c.)*

(d) DC Power Systems. Battery systems are critical to turbine emergency overspeed trip systems. Test and maintain battery systems on a regular basis to ensure system operation. Use of failsafe type systems is preferred for emergency trips (i.e., where a battery system falls below a specified energy level the powered device will fail to the safe position, tripping the turbine).

(e) Emergency Stop Valves. Stroke test all associated trip valves daily to ensure movement. Dismantle, inspect and refurbish these valves at least every five years to the extent necessary to ensure proper valve operation. Where conditions necessitate, because of abnormal wear or deterioration, more frequent dismantle activities may be necessary.

FM Global feels that electronic overspeed systems are the preferred type and recommends that they be used on units that are being phased into longer dismantle periods. Additionally, to avoid the problems inherent with mechanical overspeed systems, replacement with redundant electronic overspeed is recommended.

2.2.3 Testing of Auxiliary and Emergency Lube Oil Systems

Test the standby auxiliary lube oil pump weekly by lowering pressure to its actuating sensor, verifying that it will start at the proper pressure and that its output is consistent with the manufacturer’s specifications. If a separate emergency lube oil pump is provided, test it weekly and prior to every startup. Correct and verify by an additional test any deviations from manufacturer’s set points.

Record start and stop pressures of auxiliary and emergency lube oil pumps.

Where gravity-type emergency lube-oil systems are used, test the tank low-level alarm weekly.

Conduct annual flow test for units so equipped.

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2.2.4 Lube Oil System Management

The function of lube oil in turbine reliability cannot be overemphasized. An effective monitoring program of a lube oil system condition must be in effect. An acceptable program is based on written documentation setting forth goals and requirements that are acceptable to the manufacturer for the machine application, turbine history, and the risk.

The basic elements of a lube oil management program include but are not limited to the following:

1. Purchase specifications prepared by the plants' engineering department. Such specifications to be included with every purchase order for new oil.

2. To prevent contamination, oil storage to be in a clean, controlled environment.

3. Storage of oil in properly identified sealed containers.

4. Sampling of oil ***prior*** to use to ensure that it is the specified oil and not contaminated.

5. Oil reservoir pre-closure inspection and sign-off to prevent debris from entering the oil system following any maintenance work and following refill.

6. Leak testing all liquid heat exchangers (oil coolers) to ensure that water has not entered the system.

7. Using a qualified lab, do an oil analysis two to four times annually, depending on operating conditions and history. Additionally, conduct an analysis prior to outage planning to obtain information pertinent to the outage.

8. If oil is to be recycled on-site, specifications for the conditioner or centrifuge should specify the oil used, the purity required and the contaminants that could reasonably be encountered.

2.2.5 Bearing Alignment and Coupling Inspection

Check turbo-set bearing alignment at every dismantle. To align for steady-state operating conditions using cold alignment methodology, use estimated operating temperatures at pedestal to correct for temperature differentials. An acceptable alternative is the use of hot alignment at steady-state temperature.

Check bearing alignment at least every two years where a problem exists that requires periodic realignment of the bearings after alignment checks, or because there is a history of shaft fracture, cracking, or coupling distress.

It is sometimes necessary to dismantle a coupling prior to an overspeed trip test. This provides a convenient opportunity for coupling examination. Examine the spline teeth of a gear coupling for evidence of worm tracking, and clean the teeth of any sludge build-up.

2.2.6 Vibration Monitoring

Vibration monitoring is necessary to assist in the evaluation of a machine's condition. Include vibration tripping authorization and corrective action steps to follow in plant policies and procedures addressing handling of vibration issues.

Vibration signatures are needed to establish a baseline for monitoring and trending equipment performance. Establish new signatures any time an overhaul is performed, and more frequently if adjustments to alignment or balancing are made.

Calibrate all monitoring equipment at least annually. A check against a calibrated hand held portable monitor is satisfactory.

2.2.6.1 Vibration Monitoring Methods

There are two principle methods of monitoring machine vibration: fixed, and hand-held.

- *Fixed monitoring* – Hard wired, permanently installed used to monitor either continuously or intermittently. Where fixed instrumentation is used, visually check the vibration level at each point at least daily and record weekly, if automatic recording is not done or if a single entry satisfies the weekly summary performance log requirement vs. a reference to a strip chart, etc. Compare results and trend data weekly. Alarms should sound in a constantly attended location.

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- *Hand held* – Manual intermittent monitoring of vibration. Where this method is used, take readings at each point, record results and trend data at least daily. Take readings at the same location points to ensure consistency of data.

If monitored value differs from visual check, evaluate the trend and then investigate. Take corrective action as necessary.

Use the vibration monitoring procedure described in Data Sheet 17-4, *Monitoring and Diagnosis of Vibration in Rotating Machinery,* for turbines ≥20 MW capacity. Table 1 of Data Sheet 17-4, Section 2.1 provides guidance for diagnosis of vibration changes in steam turbines of all sizes, and presents the most likely causes of a given vibration symptom and suggests the most efficient investigative approaches.

2.2.7 Feedwater-Heater Level Controls

Failure of a level switch to respond to high condensate level can, under certain conditions, result in water induction into a turbine. For feedwater heaters with this potential, test and maintain the high-level and high-high-level switches using a program of testing similar to that described below. Periodically disassemble and clean the level switches. In most instances the level switches are the same or similar in construction and operation to a boiler low water fuel supply cutout.

Control of condensate level in feedwater heaters is usually by means of level controllers mounted on the shell or on a piping assembly containing controllers and switches that is mounted on the shell. Simulation of a high water level condition is feasible by means of a two-way valve (try cock) or the isolation and vent valves on the steam side of the switch. The steam chamber can be isolated from the heater and, at the same time be vented to atmosphere, reducing the pressure above the condensate in the gauge, and permitting the condensate level to rise and actuate the level switch at alarm level.

Test high and high-high level switches at least quarterly using the simulation method. Repair or replace switches that have inconsistent test results.

2.2.8 Calibration of Instruments

Calibrate instrumentation associated with protective and control devices on an annual basis. Verify and record alarm and trip settings.

2.2.9 Dismantle Inspections

Complete dismantle inspections involve removal of the top casing half and the rotor, or, for barrel type turbines, removal of the head flange and pulling the rotor. Turbine dismantle intervals depend on the boiler water treatment system, turbine inlet steam purity, monitored performance, turbine vibration, and the previously evaluated condition of materials in high stress areas.

The scope of a steam turbine dismantle inspection can include any or all of the following:

- Lifting turbine casings and inspecting top and bottom halves or, removing the head flange and internal casing inspection for barrel type turbines.
- Removal of the rotor(s); cleaning or grit-blasting as necessary and performing a chemical analysis of any deposits found on the rotors.
- Inspection of the steam chest and nozzle blocks.
- Removal of the nozzle diaphragms; grit-blasting if necessary.
- Nondestructive examination (NDE) of nozzle vanes for any indications.
- Weld repair of cracks, blending of nicks and dents, straightening of nozzle vane leading and trailing edges.
- NDE of blades, shrouds, and blade slots in disks or spindle.
- NDE of disks and rotors.
- Dismantle, inspection, and checking of clearances of journal and thrust bearings.
- Removal and checking of all control valve components (valve disks, seats, stems, guide, linkages, etc.). Exercising the valves does not provide ensurance that the valve disk is not broken or warped to the extent that it will not seat securely.

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- NDE of bolts and studs.
- Dismantle, inspection, and checking of operation of emergency stop valves, intercept valves and non-return valves in extraction lines. NDE of stems, valve disks and seats. Refurbishing as necessary.
- Inspection of steam strainer(s).
- Borescope examination (at fixed interval) of rotors. (Any rotor determined to be manufactured prior to 1960 should not exceed 5 year interval for borescope examination.)
- Dismantle, inspection, and refurbishing overspeed trip system.
- Checking of lube-oil system components.
- Dismantle and inspection of main, auxiliary, and emergency lube-oil pumps and checking for proper operation.
- Checking of clearances between stationary and moving parts.
- Inspection of steam glands, seals and packing.
- Inspection of oil seals.
- Alignment of bearings.
- Coupling alignment.
- Overspeed test after re-assembly.

2.2.9.1 Steam Turbines Driving Generators

FM Global policy is to establish turbine dismantle inspection intervals for steam turbines using on a time – based approach (boiler-water treatment system, purity of the steam entering the turbine and the monitored performance and vibration of the turbine). When additional operational and physical data is available and evaluated, a condition–based. (Condition-based includes time-based approach plus physical analysis.) approach may be used to establish the inspection interval, the objective being a longer interval. The dismantle interval identified for the turbine does not apply to the driven object.

Where performance, vibration and steam purity monitoring programs are not in effect for a given steam turbine, and no reasonable alternative to these programs exists, dismantle the turbine within one year after installation; or following installation of a new or redesigned rotor and subsequently, at intervals not exceeding a maximum of five years for industrial units and three years for commercial units and units that do not have steam purity monitoring.

2.2.9.2 Dismantle Intervals — Base-Loaded Units

Table 2 provides guidelines for making recommendations for dismantle (lifted-cover) inspections of base-loaded steam turbines.

Base-loaded turbines are defined as units that have no more than five cold starts per year, even though they may be on cycling duty. Peaking units require special investigation, assessing the thermal gradients developed over each cycle of operation and the degree to which the unit cools off during out-of-service periods. OEM-specific recommendations should be followed in calculating and establishing intervals.

Except for certain types of steam turbines, Table 2 recommends that under-cover inspections be based on a condition-based monitoring system. The table should be used with considerable judgment. For the high-pressure (HP) and intermediate-pressure (IP) cylinders of large turbines with the most advanced water treatment plants, the highest level of steam purity as sampled directly, a thorough in-place performance-monitoring system, and an adequate vibration-monitoring system, no specific recommendation for a fixed interval is made. However, based on present technology, a ***15-year maximum*** is considered justifiable.

For the low-pressure (LP) cylinder(s) of such a turbine, a 10-year interval is recommended. This may be exceeded if access to the L-0 and L-1 stages for visual inspection is possible without a dismantle, and if regular inspection, including nondestructive examination (NDE) of any accessible blade surfaces, has revealed no indications of cracking or other deterioration.

Solid-particle erosion of blading is a problem for turbines with steam generators operating at high pressures where steam temperatures are greater than 1000°F (538°C). It is caused by exfoliation of the boiler tubes

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and usually results in a loss of efficiency and in some cases, a loss of flow capacity. Performance monitoring and trending should be able to detect this erosion. A recommendation for a ***maximum interval*** of ten years is made for once-through boilers because they do not have a drum in which solids may be collected and blown down. However, if the water treatment and the resulting steam purity are not up to the current level of technology, depending on the degree of deficiency, fixed dismantle intervals of less than ten years may be necessary.

Table 2 provides some examples of such reduced intervals based on water treatment and steam-purity deficiencies. Any mutually agreeable inspection interval may be set, regardless of what is recommended in the table, depending on the results of performance monitoring and the degree of deterioration found in previous inspections.

Certain conditions, such as excessive steam erosion of blades or boresonic indications disclosed during a previous dismantle, might warrant a reduction in the recommended inspection interval. Performance monitoring might also reveal deposit buildup on blades, nozzle vane deterioration or labyrinth seal deterioration.

When monitored conditions indicate that a dismantle of a given turbine section casing is advisable, only that cylinder need be opened unless, after opening, it becomes evident that downstream damage may have occurred.

Nothing in this standard is to be regarded as taking the place of a manufacturer's warranty inspection, if the owner has contracted for such an inspection.

Table 2. Intervals and Conditions for Dismantle Inspections of Steam Turbines[1]

			Water-Treatment				
Dismantle Interval[2]	*Turbine Description and Rating*	*Turbine Construction*	*Boiler Type and/or Pressure*	*Make-up*	*Condensate Polishing*	*Boiler-Water*	
						Oxygen Scavenging	*pH Control*
15 Years (Suggested) Maximum	80 MW +	2 Cylinders	Drum2000 psi + (13.8 MPa +)	H_2-Cation OH-Anion Demineralizer (+ Mixed Bed)	Mixed-Bed	Hydrazine	Ammonia
10 Years (See comment on previous pg.)	80 MW +	1 or 2 Cylinders	Drum	H_2-Cation OH-Anion Demineralizer (+ Mixed Bed)	Mixed-Bed	Hydrazine	Ammonia
	LP-Cylinder		2000 psi + (13.8 MPa +)				
10 Years	80 MW +	2, 3, or 4 Cylinders	Once-Through	H_2-Cation OH-Anion Demineralizer (+ Mixed Bed)	Mixed-Bed	Hydrazine	Ammonia
	80 MW + HP-, IP-Cylinders	2 Cylinders	Drum-2000 psi + (13.8 MPa +)	H_2-Cation OH-Anion Demineralizer	None	Hydrazine	Congruent Phosphate Na: PO_4 = 2.6
7 Years	80 MW + LP-Cylinder	1 or 2 Cylinders	Drum-2000 psi + (13.8 MPa +)	H_2-Cation OH-Anion Demineralizer	None	Congruent Hydrazine	Phosphate Na: PO_4 = 2.6
	80 MW +	2, 3, or 4 Cylinders	Once-Through	H_2-Cation OH-Anion Demineralizer	None	Hydrazine	Ammonia
7 Years Maximum	< 80 MW and All Turbines in Industrial Plants			H-Cation OH-Anion Demineralizer	None	Hydrazine	Congruent Phosphate Na: PO_4 = 2.6

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Water-Treatment							
Dismantle Interval[2]	*Turbine Description and Rating*	*Turbine Construction*	*Boiler Type and/or Pressure*	*Make-up*	*Condensate Polishing*	*Boiler-Water*	
						Oxygen Scavenging	*pH Control*
5 Years	< 80 MW and All Turbines in Industrial Plants			H-Cation OH-Anion Demineralizer or Hot Lime/ Sodium Zeoplite or Soda Ash	None	Sodium Sulfite	Phosphate Hydroxide (40 ppm Phos. Max.) or Chelant
5 Years Maximum	All Turbines < 50 MW Shrunk on Wheels			H-Cation OH-Anion Demineralizer or Hot Lime/ Sodium Zeoplite or Soda Ash	None	Sodium Sulfite	Phosphate Hydroxide (40 ppm Phos. Max.) or Chelant
	All Pre-1960 Turbines and with Hollow Rotor Bores		Drum	–	–	–	–
3 Years Maximum	All Turbines in Comm'l Locations[4]			H-Cation OH-Anion Demineralizer or Hot Lime/Sodium Zeolite Or Soda Ash	None	Sodium Sulfite	Phosphate Hydroxide (40 ppm Phos. Max.) or Chelant

Steam Purity Monitoring								
Dismantle Interval	*Turbine Description and Rating*	*Turbine Construction*	*Boiler Type and/or Pressure*	*Sampling*	*Sodium*[3]*, ppb*	*Silica*[3]*, ppb*	*Cation Conduct.*[3] *mho/cm (S/cm)*	*(or) Chloride Ions,*[3] *ppb*
15 Years (Suggested) Maximum	80 MW +	2 Cylinders	Drum 2000 psi +(13.8 MPa +)	Steam	5	10	0.2	5
10 Years (See Comment)	80 MW + LP-Cylinder	1 or 2 Cylinders	Drum 2000 psi + (13.8 MPa +)	Steam	5	10	0.2	5
10 Years	80 MW +	2, 3, or 4 Cylinders	Once-Through	Feedwater	2	5	0.2	5
	80 MW + HP-, IP-Cylinders	2 Cylinders	Drum-2000 psi + (13.8 MPa +)	Steam	10	20	0.2	5
7 Years	80 MW + LP-Cylinder	1 or 2 Cylinders	Drum-2000 psi + (13.8 MPa +)	Steam	10	20	0.2	5
	80 MW +	2, 3, or 4 Cylinders	Once-Through	Feedwater	5	10	0.2	5
7 Years Maximum	< 80 MW and All Turbines in Industrial Plants	—	—	Steam	< 10	< 20	< 0.2	< 5

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Steam Purity Monitoring								
Dismantle Interval	*Turbine Description and Rating*	*Turbine Construction*	*Boiler Type and/or Pressure*	*Sampling*	*Sodium[3], ppb*	*Silica[3], ppb*	*Cation Conduct.[3] mho/cm (S/cm)*	*(or) Chloride Ions,[3] ppb*
5 Years	< 80 MW and All Turbines in Industrial Plants	—	—	Steam or Feedwater	< 10	< 20	< 0.2	< 5
5 Years Maximum	All Turbines < 50 MW with Shrunk-on Wheels	—	—	Steam or Feedwater	< 10	< 20	< 0.2	< 5
	All Pre-1960 Units		Drum	—	10	20	0.2	5
3 Years Maximum	All Turbines in Comm'l[4] Locations			Steam or Feedwater	< 10	< 20	< 0.2	< 5
	Turbines with no Monitoring[5]			None	–	–	–	–

Performance/Vibration Monitoring						
Dismantle Interval	*Turbine Description and Rating*	*Turbine Construction*	*Boiler Type and/or Pressure*	*Performance-Monitoring IS = Isentropic Efficiency F_1 = Flow-Function*	*Vibration Monitoring*	*Comment*
15 Years (Suggested) Maximum	80 MW +	2 Cylinders	Drum 2000 psi + (13.8 MPa +)	Daily-IS and F_1 of Each Shell Calculated	On-line Instruments Daily Check Weekly Recording	No Recommendations For Fixed Interval
10 Years (See Comment)	80 MW + LP-Cylinder	1 or 2 Cylinders	Drum 2000 psi + (13.8 MPa +)	Daily-IS and F_1 of Each Shell Calculated	On-line Instruments Daily Check Weekly Recording	May be increased under certain conditions[6]
10 Years	80 MW +	2, 3, or 4 Cylinders	Once- Through	Daily-IS and F_1 of Each Shell Calculated	On-line Instruments Daily Check Weekly Recording	Maximum Interval
7 Years	80 MW + HP-, IP-Cylinders	2 Cylinders	Drum- 2000 psi + (13.8 MPa +)	Daily-IS and F_1 of Each Shell Calculated	On-line Instruments Daily Check Weekly Recording	Reduced Interval
	80 MW + LP-Cylinder	1 or 2 Cylinders	Drum- 2000 psi + (13.8 MPa +)	Daily-IS and F_1 of Each Shell Calculated	On-line Instruments Daily Check Weekly Recording	Reduced Interval
	80 MW +	2, 3, or 4 Cylinders	Once- Through	Daily-IS and F_1 of Each Shell Calculated	On-line Instruments Daily Check Weekly Recording	Reduced Interval
7 Years Maximum	< 80 MW and all Turbines in Industrial Plants			Calculation of IS and F_1	On-Line Instruments: Daily Check; Weekly Recording	Minimum Requirements
5 Years	< 80 MW and all Turbines in Industrial Plants			Weekly Calculations of IS and F_1	On-Line Instruments: Daily Check; Weekly Recording	Reduced Interval

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Performance/Vibration Monitoring						
Dismantle Interval	*Turbine Description and Rating*	*Turbine Construction*	*Boiler Type and/or Pressure*	*Performance-Monitoring IS = Isentropic Efficiency F_1 = Flow-Function*	*Vibration Monitoring*	*Comment*
5 Years Maximum	All Turbines < 50 MW with Shrunk-on Wheels	—	—	Weekly Calculations of IS and F_1	On-Line Instruments: Daily Check; Weekly Recording	—
	All Pre-1960 Units		Drum	Weekly Calculation of IS and F_1	On-line Instruments Daily Check Weekly Recording	Include Boresonic Inspection
3 Years Maximum	All Turbines in Comm'l[4] Locations			Weekly Calculation of IS and F_f	On-Line Instruments: Daily Check, Weekly Recording	–
	Turbines with no Monitoring[5]			None	None	First Inspection 1 Year

Notes:

1. The above are guidelines to assist in establishing dismantle inspection intervals. Any appropriate lower interval may be set, regardless of what is recommended in this table, depending on the results of performance monitoring and the degree of deterioration found in previous inspections, and evaluations of remaining life of highly stressed components.

2. Separate dismantle intervals are required for controls and protective devices. Such intervals are discussed elsewhere in the data sheet.

3. Occasional excursions of double these limits for a 2 week period, as well as up to four times these limits for 24-hr periods are tolerable, provided steps are always taken to correct the problem promptly.

4. Commercial locations are defined as: schools, apartment buildings, and non-industrial properties of that type.

5. Applicable to turbines for which steam-purity, vibration and performance monitoring are not in effect, and no reasonable alternative exists. For these units, make a first inspection at 1 year with subsequent inspections at 3 years.

6. If access can be obtained to the L-0 and L-1 blade rows for visual and nondestructive examinations, and if regular inspection gives no indication of cracking, the maximum interval may be extended to 15 years.

Note: Once-through boiler water treatment and steam purity may be different for sub-critical, super-critical and ultra super-critical boilers. This may have a bearing on the intervals. For instance, oxygenated water treatment has been successfully applied to once-through boilers in Europe, Japan and a few in the U.S.

Do not base dismantle frequency solely on time-based monitoring. An acceptable boiler-cleaning program must be in place for conditions to be acceptable. This cleaning program may be at regular intervals or based on need as determined by a sampling of internal deposits. See Data Sheet 12-17, *Watertube Boilers*). Industry experience indicates that performance monitoring will detect the source of or the result of solid particle erosion of blades in time to prevent failures.

2.2.9.3 Evaluation of Boiler Water Treatment Plants

Pay particular attention to locations where chlorine and sodium can enter the boiler water. An anion-exchange demineralizer is needed between points where chlorine is added (including the condenser cooling water) and the boiler drum.

Faulty regeneration of demineralization resins can also result in the introduction of sodium and/or chlorine into the boiler water. Only steam purity monitoring can detect this. Sodium may be introduced directly as sodium hydroxide in coordinated or congruent phosphate treatments. The sodium-to-phosphate ratio is used to control the feed at varying flows.

For oxygen control sodium sulfite is less desirable than catalyzed hydrazine because it combines to form sodium sulfate or hydrogen sulfide. The former may deposit out on blades and vanes, clogging steam passages, while the latter may cause stress corrosion of spindles or disks.

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2.2.9.4 Evaluation of Performance

Plots of isentropic efficiency and of flow capacity over a period of time may exhibit a good deal of scatter. This may be true even though all points are taken, as they should be, at a valve-wide-open (VWO) condition or at some constant percentage of valve opening (valve position, or valve point – typically performance is predicted at control valve(s) wide open). For instance, if there are 4 control valves and all 4 are 100% open, it's referred to as VWO; 1 of 4 valves 100% open is not VWO. It is necessary to plot a sufficient number of points to allow for all the variables of instrumentation accuracy, as well as for accuracy of reading and recording. It is usually possible to develop a reliable trend. Baseline performance is defined as that level of flow capacity or efficiency that has been established for a cylinder over a period of time at a given stop-throttle or control valve opening. Hard and fast rules cannot be given for permissible levels of performance deterioration because failures have not yet been correlated with actual measured levels of deterioration.

Performance should be monitored against an as-built thermal kit. The thermal kit includes predicted thermal performance of the cycle including stage conditions at various loads. Temperature, pressure, enthalpy, etc. are plotted on a Mollier diagram.

Until such time where correlation between failure and deterioration has been developed, the following guidelines apply:

a) A deterioration in efficiency or in flow capacity of 5 to 10 percent below baseline values in any one cylinder warrants action to determine the cause. If possible, perform a wet steam internal wash of the cylinder in case there are water-soluble deposits on the blading.

b) If washing does not correct the problem, and both efficiency and flow capacity continue to be low, the cause is probably hard deposits of non-reactive silica or severe deterioration of nozzle vanes.

c) If only efficiency is affected, the cause is probably damage to or deterioration of interstage labyrinth seals (packings) or severe blade erosion.

d) In case of either b or c, schedule a turbine dismantle inspection within 12 months to determine the cause and correct it.

e) Deterioration should never be allowed to exceed ten percent of baseline values.

2.2.9.5 Cyclic, Two-Shift and Peaking Duty

Perform periodic dismantle inspections based on the numbers of cycles of various types (cold starts, warm starts, two-shift operations and hot starts) that are accrued, as well as to some extent on the acceleration and load-ramping control systems employed. Only the manufacturer (or in some cases, specialist consultants) can establish the low-cycle fatigue life (LCF) and the permissible fracture-mechanics cycles (FMC) for a turbine rotor. This should always be done for a unit in this service.

Make a dismantle inspection of a turbine cylinder after a number of cycles corresponding to between 80 and 100 percent of the LCF life of the rotor, as determined by the above-mentioned methods.

Include nondestructive examination of the turbine rotor surface, including blade retaining slots over their complete axial length and the rotor bore, if so designed, using ultrasonic techniques (boresonic inspection). Any patterns of indications that could be cracks should be repaired by qualified personnel in accordance with the manufacturers procedures or procedures approved by the manufacturer. Manufacturers provide limits of radial material that may be removed without weakening the shaft excessively.

Conduct subsequent dismantle inspections at intervals corresponding to the fracture mechanics limits as determined by engineering analysis.

2.2.9.6 Dismantle Intervals — Mechanical-Drive Steam Turbines

Refer to Table 3 for recommendations for dismantle inspection of mechanical-drive steam turbines.

Mechanical-drive steam turbines are used in commercial complexes, schools and universities, and apartment buildings to drive refrigerating and air conditioning compressors and in industrial locations to drive generators, paper machines, pumps, fans and blowers, etc., in addition to air conditioning compressors.

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Steam may be obtained from a large process boiler or a packaged boiler, or may be purchased from a local utility. In some cases the operator may have little control over the purity of the steam, and little capability for monitoring steam purity, performance and vibration. In addition, maintenance may be by contracted to an outside firm.

Dismantle turbines operating under the these conditions on a fixed schedule with intervals not exceeding three years.

Recommendations for dismantle inspection intervals are identified in Table 3. This activity should be in addition to the manufacturer's warranty inspection (if performed).

Table 3. Summary of Dismantle Intervals for Mechanical-Drive Steam Turbines[1]

Type of Occupancy	*Maximum Interval*
Electric Generating	7 years
Industrial Plant	7 years
Steam Turbines of <50 MW (67,000 hp) having shrunk-on rotor wheels	5 years
Commercial Property	3 years
All Occupancies:	
Steam-purity, vibration, and performance monitoring not in effect, and no reasonable alternative to monitoring in place.	1 year (first inspection) 3 years (subsequent inspections)

Notes: Extended dismantle frequency may be justifiable based on condition assessment.

1. Both industry and FM Global loss experience show that the following types of failures develop as a result of internal conditions in a steam-turbine casing:

- High frequency fatigue fractures of blades and blade shroud bands
- Corrosion fatigue cracking of blades in last stages
- Stress corrosion cracking of disks or spindles (at rim or bore)
- Some thrust bearing wiping
- Loosening of disks on shafts (in shrunk-on disk construction)
- Loosening of reversing-blade blocks in Curtis stages

Dismantle inspections are of limited benefit in reducing the first four of these types for many industrial turbines. For such turbines, controlling and detecting the conditions that could lead to a failure are more effective loss prevention techniques. In the case of the last two failure modes, more frequent dismantle inspections have been recommended in Table 2 for units that might be susceptible to such hazards.

2.2.10 Summary of Loss Prevention Recommendations

Table 4 is a summary of the above recommendations for loss prevention systems and testing. The concept of ''Advisory Practices'' is briefly described in Section 2.1.1. These recommendations pertain to all turbines, as applicable.

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Table 4. Summary of Loss Prevention Recommendations

Overspeed, Water Induction and Lube Oil	
Exercise emergency stop valve and trip mechanism	Weekly
Exercise extraction non-return valves	Weekly
Test emergency stop valve	Annually
Dismantle, inspect and refurbish turbine emergency stop and non-return valves	At least every 5 years*
Overspeed test (dependent on design)	Annually (Mechanical)
Test high and high-high level switches on feedwater heaters	Quarterly
Dismantle and clean level switches on feedwater heaters	Every 3 years
Test steam traps or power operated drain valves in turbine steam admission lines	Monthly
Test auxiliary and emergency lube systems	Weekly
Bearing Alignment	
[No history of Problems]	At every dismantle
[With history of misalignment]	Every two years
Vibration Monitoring	
With installed instrumentation, visual verification	Daily
With installed instrumentation, log readings	Weekly**
Manual instrumentation, log readings	Daily
Calibrate instrumentation	Annually

* Assuming an acceptable condition based inspection/maintenance program is employed that includes evaluation of highly stressed critical components; otherwise more frequent intervals are justified. If a turbine is dismantled at more frequent intervals, then overspeed trip system should be overhauled with the turbine.

** Weekly unless data archiving is provided.

2.3 Alert

2.3.1 Schutte & Koerting Swing Check Valves

2.3.1.1 Purpose

To advise of past operating problems encountered with the operation of the subject valves.

2.3.1.2 Background

The following applies to Schutte & Koerting valves from 3 in. (8 cm) through 48 in. (1.2 m) in size used in steam extraction/non-return service on steam turbines.

Failure of the subject valves to fully close was first brought to our attention in 1985. At that time we were advised of a valve modification that involved a change in the method of securing the disk nut to its shaft This modification had been issued by the manufacturer in November 1976.

The problem involved failure of the carbon steel cotter pin that secured the disk nut to the disk shaft Failure of the pin allowed the nut to back off sufficiently, tilting the disk and preventing proper seating. This allows a backflow of cold steam causing vibration and an overspeed potential.

To prevent the disk from loosening or separating from the tail link the manufacturer recommends the following modifications be made to all S&K swing check valves:

- Replace the cotter pin with a stainless steel dowel pin, bent at both ends to prevent its coming out of the nut/shaft (See Fig. 1, top view.)
- Install an anti-rotation pin between the tail piece and the disk to prevent rotation of the disk in the steam flow path. (See Fig. 1, side view.)
- Weld the nut to the disk with a $\frac{1}{8}^{th}$-in. (3 mm) seal weld. (See Fig. 1, Revision block.)

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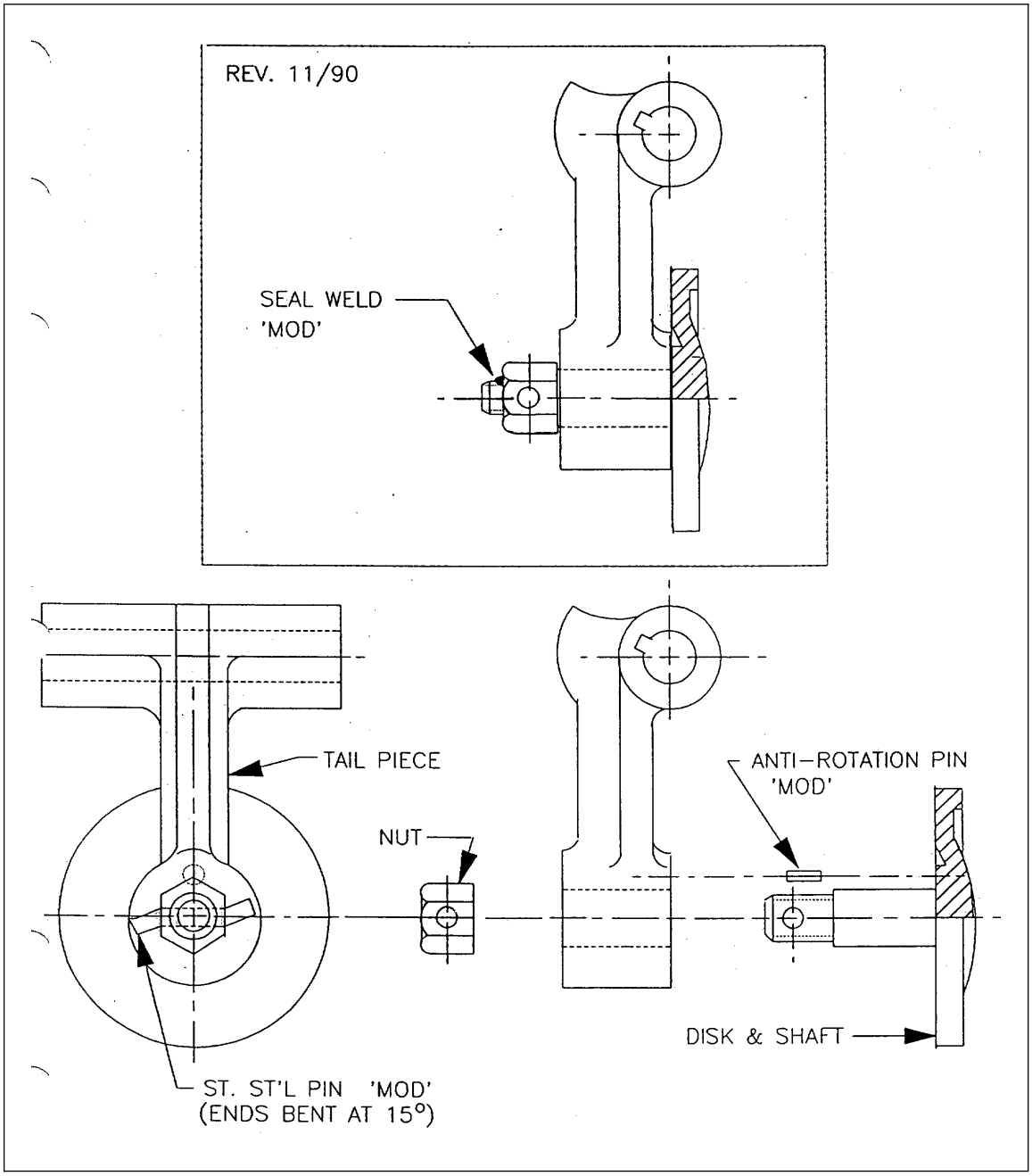

Fig. 1. Schutte & Koerting Swing check Valve Showing Modifications (Marked 'MOD').

Need for an anti-rotation device became apparent in an incident at a paper mill involving a 14 in. (35 cm) Schutte & Koerting air assisted extraction/non-return valve on a turbo-generator.

During a planned shutdown, the turbine was overspeed trip tested and all appeared to be normal, except that it was noted that after the trip, turbine speed was increasing. The speed was controlled manually and the unit was shutdown, but not before the speed had exceeded 4900 rpm.

Investigation revealed that the disk had separated from the tail piece and fallen to the bottom of the valve, allowing cold steam to back-feed into the turbine causing an overspeed condition which resulted in damage to both the turbine and generator.

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The cause of the incident was determined to have been wearing away of the nut and pin by rotation of the disk until the nut came free of the shaft allowing the disk to fall off.

Rotation was such that the disk shaft bore in the tail piece was worn elliptically to almost twice its normal size. This valve had not been fitted with the anti-rotation device as had been recommended by the manufacturer.

2.3.1.3 Recommended Action

Determine whether:

1. the plant has valves of this type in extraction/non-return service on their turbines; and

2. the discussed design changes have been implemented.

The plant may obtain details for the modifications outlined by contacting Schutte & Koerting, 2233 State Road, Bensalem, PA 19020, (215) 639-0900.

2.3.2 Inspection Interval Support for Overspeed Protection Components

This data sheet contains a recommendation that the emergency overspeed trip systems for steam turbines be completely inspected at least every five years. This is a significant departure from earlier standards in which it was assumed that appropriate inspections would be made at the time of turbine dismantle. Therefore, it is believed that an outline of the thinking that went into this recommendation would be helpful to field personnel who must consider the interval requested by the customer and the basis presented against the previous interval that was based on mechanical bolt systems. This thinking was as follows:

1. FM Global believes prevention of overspeed is the most critical loss prevention item in connection with steam turbine operation.

2. Inspection of emergency stop valves and overspeed trip systems is not always performed at the time of dismantle inspection of steam turbines. Contractors, including the original equipment manufacturers, do this inspection ***only*** when it is specified in the statement of work given them by the steam turbine owner at the time of bidding on the job. They may sometimes recommend that it be included as a critical item, but frequently, under the pressure of competition, they do not.

3. Nonreturn valves in extraction lines are frequently neglected during turbine dismantles. Sometimes they are not considered part of the turbine system. Excessive reliance is placed on exercisers that merely indicate that the valve-disk shaft will rotate, at least part way. Exercising does nothing to provide assurance that the valve disk is not broken or warped to the extent that it will not seat securely, nor that the valve seat is not severely eroded and unable to seal properly.

4. Significant improvement has been made by industry in the design and reliability of emergency overspeed trip systems. Since they were introduced in the early 1980s, electronic systems, with a 2 out of 3 voting logic capability, have become the standard offering by all major steam turbine vendors. Frequent testing of electronics and exercising of emergency stop valves can ensure operability. Some vendor systems can provide simulated off-line testing that may negate the need to perform actual overspeed testing through the use of frequency generators.

Inspection intervals for mechanical bolt systems remain as an annual requirement. Inspection intervals for electronic systems are at least every three years with more frequent intervals required if operability-testing features are limited.

Inspection/dismantle intervals for emergency stop and non-return valves has been extended to at least every five years. The potential for internal wear, distortion or foreign object damage that could compromise leak tightness or parts failure should be monitored and more frequent intervals may be required.

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3.0 SUPPORT FOR RECOMMENDATIONS

3.1 Loss History

Tables 5, 6 and 7 provide the loss history by failed part for FM Global-insured utility, industrial and mechanical drive turbines.

Table 5. Utility location steam turbine loss experience (1984–present)

Damaged Part and Type of Damage	*Percent Losses*	*Percent Gross U.S. Dollars*
Blading		
Fracture[1]	24	23
Overheating	<1	5
Rubbing	4	4
Loose Part Damage	5	3
Erosion and Corrosion	6	1
Overspeed		
Operator Error	<1	2
Trip Malfunction	<1	10
Shaft Cracking	6	7
Wheel Cracking and Burst	3	9
Shroud Band Separation	5	5
Bearing/Journal		
Seizure (Lack of Lube Oil)	4	3
Seizure (Other than Lube Oil)	4	4
Journal Pitting and Scoring	1	1
Nozzle Block Cracking	1	2
Coupling Cracking	1	2
Rotor Shaft Bowed or Cracked	2	2
Extraction/Check Valve	1	2
Steam Strainer	1	<1
Casing Cracking	<1	<1
Miscellaneous (Not Specified)	11	7
Unknown	16	10

Note: 1. Includes blade failures due to high frequency and corrosion fatigue.

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Table 6. Industrial steam turbine losses. (1984–present)

Damaged Part and Type of Damage	*Percent Losses*	*Percent Gross U.S. Dollars*
Blading		
Fracture	10	17
Distortion	<1	4
Carryover Damage	<1	2
Rubbing	5	10
Loose Part Damage	3	3
Shroud Separation	3	2
Shaft Cracking	3	2
Wheel Cracking, Deforming	4	4
Bearing/Journal		
Seizure (Lack of Lube Oil)	12	12
Seizure (Other than Lube Oil)	11	8
Diaphragm and Nozzle Block Cracking	2	6
Coupling Cracking	>1	<1
Rotor Shaft Bowed or Cracked	5	4
Throttle Valve	>1	7
Seals	3	>1
Miscellaneous (Not Specified)	16	11
Unknown	20	6

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Table 7. Mechanical-drive steam turbine losses (1984–present)

Damaged Part and Type of Damage	*Percent Losses*	*Percent Gross U.S. Dollars*
Blading		
Fracture	10	25
Distortion	<1	3
Loose Part Damage	2	2
Shaft Cracking or Distortion	8	5
Wheel Loosening	<1	4
Bearing/Journal		
Seizure (Lack of Lube Oil)	9	5
Seizure (Other than Lube Oil)	7	7
Journal Pitting and Scoring	<1	<1
Nozzle Block Cracking	<1	<1
Coupling Cracking	2	4
Rotor Shaft Bowed or Cracked	9	2
Check or Throttle Valve	3	9
Cracked Casing	1	<1
Seals	1	<1
Bolt failure	2	3
Miscellaneous (Not Specified)	15	8
Unknown	27	14

Overspeed losses and thrust bearing failures (not related to water induction or loss of lubrication) are frequent in mechanical drive and industrial steam turbines driving generators. For the time period noted, only 3 losses involving overspeed occurred in steam turbines at utility locations which accounted for 2 percent of the total number of losses and 13 percent of the dollar value (two were turbo-generators and one was boiler feed pump).

Table 8. Overspeed losses based on occupancy

Occupancy	*Percent Losses*	*Percent Gross U.S. Dollars*
Woodworking	12	1
Paper Products	43	58
Chemical	7	3
Petro-Chemical	10	<1
Pharmaceutical	2	<1
Food Products	10	3
Gas Plant	2	<1
Electric Power	7	7
Cogeneration	2	26
Miscellaneous	5	<1

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Table 9. Overspeed Loss History by Turbine Usage

Turbine Usage	*Percent Losses*	*Percent Gross U.S. Dollars*
Turbine Driving Electric Generator at Utility Locations	5	6
Turbine Driving Electric Generator at other than Utility Locations	40	60
Mechanical-drive Turbine	55	34

3.2 Hazards to Steam Turbines

3.2.1 Overspeed

Overspeed is the most serious hazard to steam turbines, both in terms of frequency and severity. However, as previously indicated, FM Global has only minimal loss history involving overspeed incidents involving utility steam turbine-generators. A major reason for this is that utility steam turbine-generators have redundant overspeed trip systems (mechanical and electrical or electronic multi-voting).

Overspeed occurs when the driven object loses load either accidentally or by operator error. Overspeed also can occur during start-up prior to load application. When this happens, turbine speed tends to increase very rapidly if not restrained, until a speed is reached at which one or more of the following may occur:

- A turbine disk or spindle fractures due to overstress
- The generator retaining rings become loose or fracture due to overstress and come off, permitting the rotor end turns and eventually the entire field winding to fly out destroying the generator
- Turbine blade attachments deform or fracture, releasing blades or allowing them to become loose
- A turbine disk stretches to the point of being loose on a shaft
- A turbine wheel stretches to the point of severe blade rub

A number of secondary effects also are possible, such as:

- Very severe vibration due to unbalance.
- Fracture of a shaft due to overload arising from the excessive whirl of the rotor. Shaft fracture is not a primary effect of overspeed, and a fractured shaft between the steam turbine and the generator may be the cause of the overspeed. In such a case the fracture surfaces usually exhibit evidence of fatigue.
- Severe bearing damage due to rotor whip/whirl.

Steam turbines can withstand some overspeed but, since stress varies as the square of the speed, this is necessarily limited. Most manufacturers today limit overspeed design provision to eight or ten percent. Older units were set for ten or twelve percent.

In the case of a turbine-generator, the generator may lose its load by erroneous tripping of the unit breaker by an operator, or by some electrical fault that causes a relay to open the breaker in a simultaneous trip mode (generator breaker, turbine stop valve and exciter breaker). In situations where an electrical fault does not represent an immediate threat to the unit, it is good practice to use the sequential trip mode in response to relay action. This involves actuating the turbine stop valve first, and then, after a delay of roughly three seconds, actuating the generator and exciter breakers. When the stop valve reaches the end of its travel, it actuates a limit switch, which then actuates the generator breakers.

An optimum sequential trip scheme utilizes a reverse power relay in conjunction with the valve limit switch. Within seconds of the stop valve closing, this relay senses reverse power flowing back into the generator from the utility grid, proving that the turbine is actually slowing down. In this sequential trip system both the stop valve closed permissive and the reverse power permissive are needed before the generator and exciter breakers are permitted to open.

Note: Where this sequential trip system is used, it is recommended that redundant reverse power relays or a reverse power relay with a bypass switch be used to ensure system reliability.

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The purpose of a sequential trip is to maintain load on the steam turbine until the steam has been safely shut off. If, after an electrical fault requiring a trip, the turbine stop valve does not close completely, thereby actuating the generator breakers, it will be evident to the operator that the unit is not unloading. If steam block valves are provided, the operator can then take steps to have steam to the turbine shut off.

Generator motoring occurs when the steam supply to a turbine is shut off while the generator remains connected (to the grid). The lack of steam energy to overcome turbine-generator losses allows the generator to act as a motor. With an energized field it drives the turbine as a synchronous motor, which can cause turbine hood and last stage blade overheating due to lack of steam flow (loss of windage) and also damage the generator (overheating, etc.). A reverse power relay can be used to detect this motoring and alarm control room operators. This relay application might not be connected to a sequential trip scheme. Historically, turbine-generator overspeed damage is more significant than that caused by motoring.

Another mechanism for sudden reduction of load on a steam turbine is shaft or coupling failure. This type of incident could be associated with a turbine driving a generator or other rotating machine. The causes and prevention of shaft and coupling cracking are discussed in Section 3.2.7. The significant consideration in these cases is that it is important to maintain the overspeed trip system in perfect working order.

Protection against overspeed is provided by the overspeed trip system, the high and low pressure stop valves, the reheat stop valves and the nonreturn valves in the extraction lines.

Such systems consist of:

1. the bolt or other device on the shaft which is caused to function by overspeed and which triggers a mechanism leading to closing of the emergency stop-valve;

2. the electronic or hydraulic system which causes the valve to function, and

3. the emergency stop valve itself.

Many things can go wrong with overspeed trip systems. If the control valves and an emergency shutoff valve fails to close completely when a turbine load is lost, steam continues to flow through the turbine, accelerating it to overspeed.

A malfunction of one (both in the case of extraction lines to the deaerator for large units) of the extraction nonreturn valves after a turbine trip would allow steam from the feedwater heater to flow back through the turbine to the condenser. This is a common cause of overspeed of steam turbines.

There are systems for testing the functioning of the electronic and hydraulic systems, and in some cases the emergency stop valve itself can be exercised while in operation to provide assurance that it is free to move. However, the only valid test for the functioning of the entire protective system is an actual overspeed test with the driven load removed. Such a test may be accomplished using a system that can send a simulated (false) overspeed signal (frequency/signal generator) to trip the turbine without actually overspeeding the unit. This method will test the system without overstressing unit components. When this type of overspeed tripping system is used, the system must be qualified by previous in-service applications with reliable documented results. Procedures must be in place to ensure that speed probes are calibrated prior to their being installed.

The overspeed trip system and the associated valves must be maintained within specified working order limits by means of regular overhaul and refurbishment activity.

3.2.2 Water Induction

The introduction of water into any part of a turbine operating at high speed and temperature can cause serious damage to the rotor, blades, thrust bearing, and interstage packing. Water induction or carryover can cause five types of turbine damage:

1. *Thrust bearing failure.* Water carryover from the steam generator may impose a load on the thrust bearing that is sufficient to cause bearing failure.

2. *Damaged blades.* Blades may be bent, or broken, as a result of high velocity impact with a large slug of water.

3. *Thermal cracking.* Water from any source may contact metal parts that are at temperatures high enough to result in thermal cracking.

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4. *Rub damage.* Water introduced from the main steam or reheat lines can cause differential expansion problems between rotating and stationary parts in the form of axial rubs. Water backing up from extraction lines may cause contraction of the lower part of the shell, giving a humping effect that can lift diaphragm packings against the rotor, causing radial rubs. Bowing of the rotor results when packing rubs cause uneven heating on the rotor surface. This additional distortion further increases the intensity of rubbing. Packings, spill strips, and blade shroud bands are the most frequently damaged parts. Water induction may cause the casing to become warped (thermal distortion) leading to subsequent rubbing. Factory heat-treating may be required to restore its shape.

5. *Permanent warping or distortion.* This condition may result when metal parts are subjected to severe quenching, and can cause steam leaks in valve and shell joints. Diaphragm dishing and rotor bowing caused by water quenching can result in distortion to the extent that turning gear motors will load and trip following engagement.

Water may be inducted into a steam turbine from three sources: the boiler (drum, superheater or reheater) carrying over into the steam inlet line, and the extraction lines to feedwater heaters and, the steam seal system. Water may be introduced by attemperators (desuperheaters) serving superheaters or reheaters. Following are techniques to protect a steam turbine from each of these sources:

3.2.2.1 Induction From the Boiler Drum, Superheater, or Reheater

a) On boiler-turbine-generator installations, a high-water level sensor on the boiler drum, set to send an audible alarm at a level that will give the operator time to control feedwater flow before induction into the turbine occurs.

b) Thermocouples in the main and reheat steam lines, installed at the top and bottom of the line set to detect a differential of 50°F (27°C) between top and bottom thermocouples, and to trip the turbines when this occurs. To detect the presence of a slug of water as early as possible, locate the thermocouples well upstream of the turbine.

3.2.2.2 Induction From Condensate in Main and Reheat Steam Inlet Lines or the Steam Seal Supply Line

Depending on unit configuration, install steam traps or drain pots with level switches or thermocouples that activate power operated valves at low points in the main steam and reheat lines, and just upstream of the emergency stop valves. Test these traps or valve systems for satisfactory operation at monthly intervals. Testing methods include:

a) *Visual,* in which the trap or pot is drained manually, and is considered satisfactory if live steam is emitted;

b) *Thermal,* in which the temperature of the trap, or of the steam lines upstream and downstream of the trap or drain pot, is observed by pyrometry or thermocouples.

c) *Acoustic,* in which the sound characteristic of the trap is noted by means of an ultrasonic tester or a stethoscope.

3.2.2.3 Induction From Feedwater Heater Extraction Lines

ANSI/ASME Standard No. TDP-1-1998, Part 1, *Recommended Practices for the Prevention of Water Damage to Steam Turbines used for Electric Power Generation*, recommends that the extraction steam system be designed so that no single failure of equipment can result in water entering the turbine. Specifically, the standard recommends that, where feedwater heaters are installed, 1 and 2 or 1 and 3 of the following three protective systems be employed in each extraction system.

System 1. A normal primary feedwater heater drain line with its associated level controller and valve and an automatically operated alternate high level drain line and valve with its level controller are illustrated in Figure 2. An alarm should accompany the activation of the high-level drain valve. The high level system protects against a faulty normal-level control system or excessive drain flow due to heater tube ruptures.

System 2. An automatic shutoff valve in the extraction line between the turbine and the heater, power-driven and actuated by high water level in the heater shell upon a high-high level signal, is shown in Figure 3. The ASME standard does not consider a check valve, whether it is of the free-swing, power-assisted, or positive closing design, to be an adequate substitute for an automatic power-driven valve, because experience indicates that seats and disks are frequently distorted.

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An additional reason for the failure to function is that a simple water backup does not produce a differential pressure in the line to close such a check valve.

System 3. Automatic shutoff valves on all sources of water entering the heater shell, implying a bypass system for the feedwater. Figure 4 shows this system in the normal operating state. In the emergency condition, feedwater flow bypasses the heater.

The purpose of the bypass system is to protect against leaking feedwater tubes. Normal operation is to be activated by a high-hi level condition.

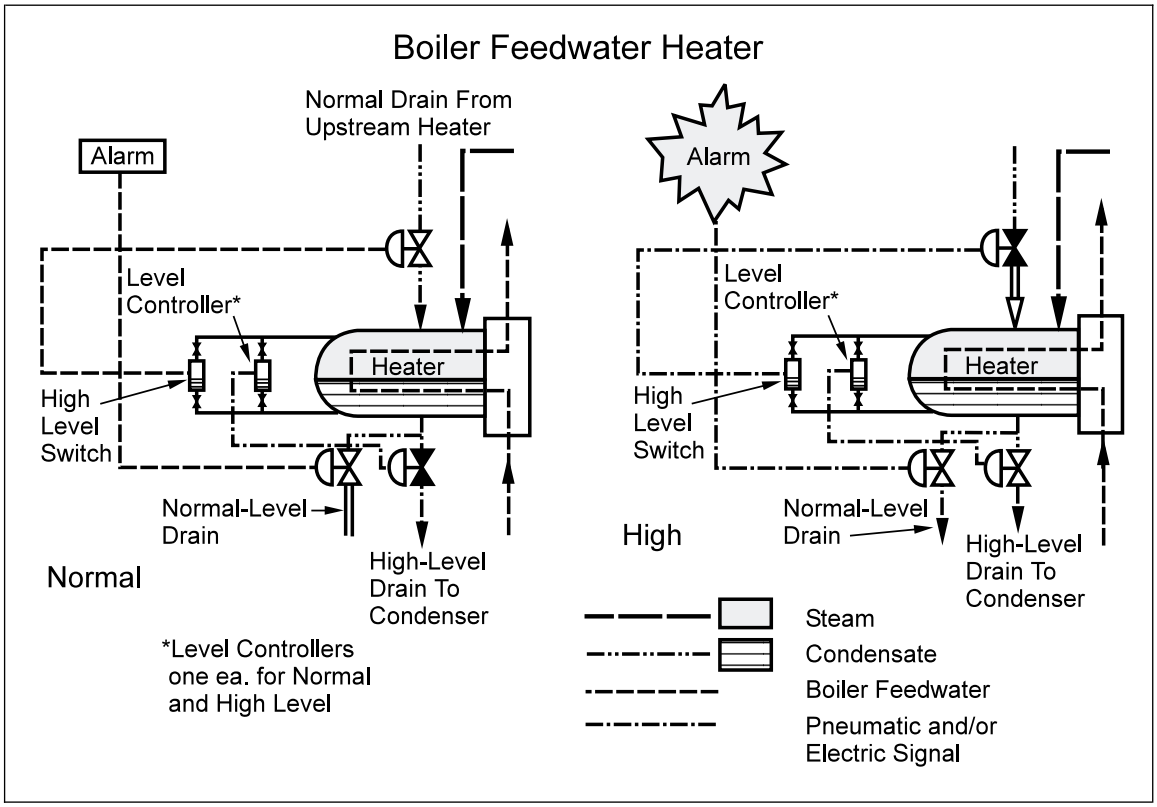

Fig. 2. High level drain on feedwater heater.

3.2.2.4 Cold Steam Backflow

When a turbine-generator is tripped, and the steam admission stop valves (main, reheat auxiliary as applicable) are closed, the pressure in the turbine drops to the vacuum maintained in the condenser. Steam in the extraction lines is prevented from expanding back through the turbine by the non-return valves in the extraction lines. If a non-return valve fails to close when subjected to the back pressure, steam from the feedwater heater serviced by that line flows back and can drive the turbine into overspeed if the generator breaker is also opened. Condensate in the heater flashes into steam close to the saturation line, and flows back into the turbine. This adds to the energy available to overspeed the turbine. However, in the case of large utility turbines, there have been incidents where the cold steam impinging on one side of the inner shell has cooled it rapidly, causing the inner shell to deform. The resulting severe rotor rub prevented the overspeed, but the blading damage was extensive.

There have been three incidents of cold steam backflow in insured utility steam turbines over a recent fifteen-year period. In each case it was determined that the nonreturn valves did not function properly because of inadequate maintenance. Meticulous maintenance of these valves is the recommended loss prevention approach.

Manufacturer's recommendations for main, reheat and extraction steam line drains must be strictly adhered to including those that are inserted into a header at the receiving vessel. This covers start-up and operating drains, the receivers they should be routed to and the sequence in which they must be installed if they are

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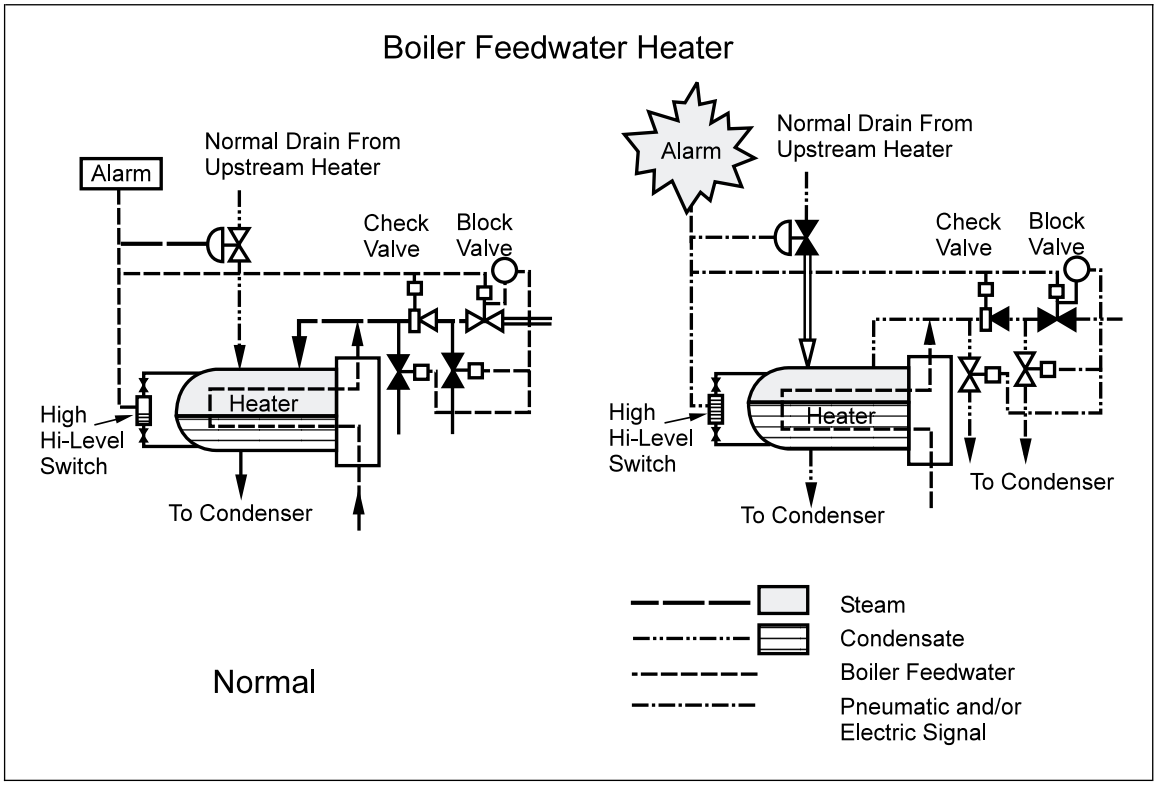

Fig. 3. Steam cutoff block valve in extraction line.

routed to the receiver (such as the condenser) via a header. Failure to follow manufacturer's recommendations has resulted in water induction and/or quenching of emergency stop valve disks that resulted in distortion or failure.

3.2.3 Blades and Shrouds in Early Stages

Most fractures of blades and shrouds in the high- and intermediate-pressure cylinders of steam turbines are caused by high frequency fatigue resulting from resonant vibration.

There are two categories of blade cracking and fracture that need to be differentiated. These are: 1) Blade fatigue (fracture of blades in the root attachment or in the blade anywhere from the root out to mid-span) and 2) Shroud and tenon fatigue (cracking and fracture of shroud bands and/or blade tenons in packeted-blade rows).

Blade Fatigue. Blade fatigue results from low order resonant vibration. Resonant blade vibration results from two concurrent circumstances. First, the blade or packet must have a resonance at operating speed, i.e., its natural frequency must be a low-order multiple (2X, 3X or 4X) of running speed, or a multiple corresponding to some set of evenly spaced obstructions (such as structural support struts) in the flow path. Second, the circumferential flow distortion must be sufficiently severe to generate vibratory stresses in the blade in excess of the fatigue strength.

The first of the above circumstances is avoided by appropriate blade and packet design, tuning out resonance at all operating speeds. The second is avoided, first in design, by attention to steam flow path details, and second, in operation, by avoiding buildup of chemical deposits on stationary surfaces in the flow path, as well as damage to stationary vanes and diaphragms. The latter is the principal approach to loss prevention. Chemical deposits are usually silica or compounds of sodium. Buildup can usually be detected by a drop in efficiency and flow capacity at a given operating condition. Thus, two powerful approaches are available to turbine operators to avoid blade fracture in early stages: control of steam purity, and detection of deposit buildup on blade and vane surfaces (as well as of possible nozzle vane damage and deterioration) through performance monitoring.

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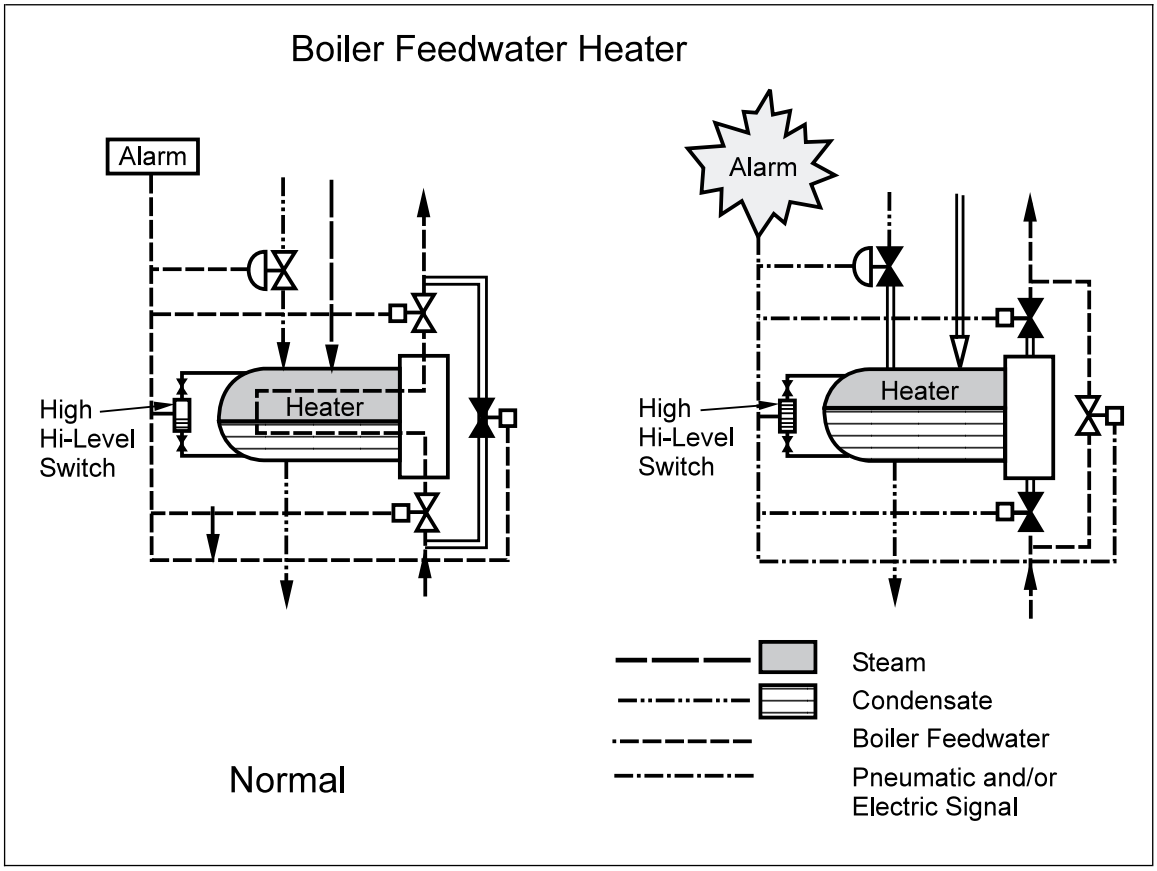

Fig. 4. Feedwater bypass section.

Development of the initial fatigue crack and progression across the blade to complete fracture is relatively rapid. Once a blade is in resonance and the excitation level is sufficient to produce stresses higher than the fatigue strength, the initial crack will appear in 10^7 cycles of vibration. This could occur in as little as 15 hours for a third-order resonance. Progression to complete fracture could then occur within a few hours. While fractures probably never occur that rapidly in practice, it is obvious that a scheduled inspection might not reveal an incipient failure.

Prevention of this type of fracture should be based on avoiding the conditions that might lead to high-frequency fatigue. These conditions can be detected as they progress, by monitoring the performance of the turbine, particularly its efficiency and flow capacity. Blade vibration, which leads to fatigue, is caused by flow distortion due to chemical deposits on stationary nozzle vanes. A less frequent cause of flow distortion is deterioration of nozzle vanes and diaphragms.

In some cases the basic turbine design itself provides sufficient flow distortion to cause fracture of blades that are in a low-order resonance. In such cases, there is nothing the user can do to avoid it, except ensuring at time of purchase that the design is mature. This type of failure tends to occur early in the life of the turbine. However, despite trouble-free operation through the early life of the turbine, it is prudent that the user still assume that some blade resonance may exist and should, therefore, monitor the performance of the turbine, as well as the purity of the steam to minimize the risk of such fractures.

Partial-arc admission is common with turbines having multiple control valves. This mode of operation results in severe distortion of the steam flow to the control stage. It is important, therefore, that control stage blades do not have low-order resonances, and to that end that they are sturdy in construction and are shrouded and banded into packets. The impact of this flow distortion may be greater on the control stage disk. There has been a major fracture of such a disk in response to partial-arc operation, and it is important that low-order disk resonances be avoided. This is the responsibility of the manufacturer. The user can do nothing to anticipate such a resonance.

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3.2.4 Shroud and Tenon Fatigue

Shroud and tenon fatigue also is a straight high frequency fatigue phenomenon resulting from nozzle vane passing order resonance. However, the frequency is much higher than in the case of blade fatigue fractures. The natural frequency of shroud edges and segments between blades is very high, and it takes a high frequency impulse to drive the resonance. This impulse is provided by the adjacent stationary nozzle vanes (of some number = N). When one of the natural frequencies of a shroud band coincides with nozzle-passing frequency (N $\times$ running speed) vibration of the band occurs, possibly with sufficient amplitude to initiate fatigue cracking.

The cracking may occur in the shroud-band supports (i.e., in the blade tenons) instead of in the band itself. This phenomenon of cracking and fracture of blade tenons has been attributed to improper peening of the tenons, and much effort has been expended by manufacturers to improve quality control in peening, both by hand and by machine. While it is true that improper assembly may have some influence on the susceptibility of a shroud band to cracking, especially if stress raisers are produced, the basic problem is still resonance, and the remedy is to design the shroud band so that it does not have such a resonance. This, however, is not easy to do, and manufacturers have frequently had to learn from experience and to try a number of modifications before a satisfactory configuration was found.

Fracture of blade tenons is basic to the design of the turbine, and the flow distortion of concern is usually not aggravated by chemical deposits. Therefore performance monitoring and control of steam purity are of very little, if any, value for its prevention. There is nothing a user can do to help prevent the initial onset of cracking.

Cracking of a shroud band frequently involves replacement of most of the entire row of blading because there is no method of removing a tangential entry blade without breaking the bands on all blades between it and the nearest blade loading slot. The damage to the row has been done. The interest in detecting this damage at an early stage comes from three considerations:

1. Identification of the problem as a design problem, the province of the manufacturer;

2. Avoidance of consequent damage downstream. There is a tendency for shroud band damage in one stage to lead to further damage downstream;

3. Avoidance of consequent damage to same stage. There have been cases in which the fracture of the shroud left blades free-standing, and these blades now were in low order resonance and vibrated to fracture in the root area, and in one known case, causing cracking of the rotor spindle.

Because of the high frequencies involved, initial cracking may occur very rapidly. The unbalance caused by broken sections of shroud may be sufficient to produce an increase in bearing vibration. Further, when the shroud bands have broken up to any great extent, the efficiency of the turbine can be affected markedly. In most cases shroud band fracture has been detected as a result of an increase in vibration. In a few cases, a drop in turbine performance or reversed rotor thrust led to the discovery. To detect shroud band damage at an early stage, vibration and performance monitoring should be used.

3.2.5 Corrosion Fatigue of L-0, L-1 and L-2 Blades

Fractures of blades in the last two or three stages of low-pressure (LP) cylinders of utility steam turbines are usually caused by corrosion fatigue. In this phenomenon, 2X, 3X or 4X resonance as described in Section 3.2.3 must be present, although resonant orders 5X and 6X also may be significant. However, in these stages flow distortion need not be as great; and it need not become aggravated. Instead, the fatigue strength of the blade material is lowered as a result of corrosive attack, usually in the form of pitting. This pitting may take place on the airfoil, in the blade tenon under the shroud band, in the area of the lacing wire, or in the root attachment area. The principal corrodent is sodium chloride, which deposits out on critical areas of the blade without any measurable dimensional buildup that can be detected by performance monitoring.

The only approach is to manage the concentration of sodium cations and chlorine ions in the steam by control of the boiler water purity. In addition, it is necessary to monitor these concentrations in the steam entering the turbine. Sodium cations are detected by continuous analysis of a condensed steam sample using the ISE (ion-selective-electrode) technique. Instrumentation is available to measure concentration of chloride ions directly, but the indirect approach of measuring cation-conductivity of condensed steam is the most commonly used technique.

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Since corrosion fatigue relies on the progress of corrosive attack, it is a slow process relative to high-frequency fatigue. However, FM Global experience, as well as an industry survey of utility steam turbines, indicates that a great majority of such fractures occurs within three years. In view of this rapid progression, it is not cost effective to rely on lifted-cover inspection as a preventive measure.

Some low-pressure cylinders have access openings to permit some inspection of the L-0, and possibly the L-1, stage of the turbine. Borescopes also can be used to examine blades in some installations. These techniques are useful because they may detect the existence of cracking in shroud bands and lacing wires. However, many instances of corrosion fatigue originate in the blade tenons under the shroud bands or in the blade fir tree attachment area. Although nondestructive testing methods have been somewhat successful in detecting actual cracks in tenons, these areas are not readily accessible for inspection. Due to the limited access, it may be difficult, if not impossible, to apply a nondestructive technique to these blades with sufficient coverage to provide a high degree of assurance when access is merely through the inspection manholes. (Bibliography Ref. 1)

Figure 5 shows the pitting that may exist under a shroud band.

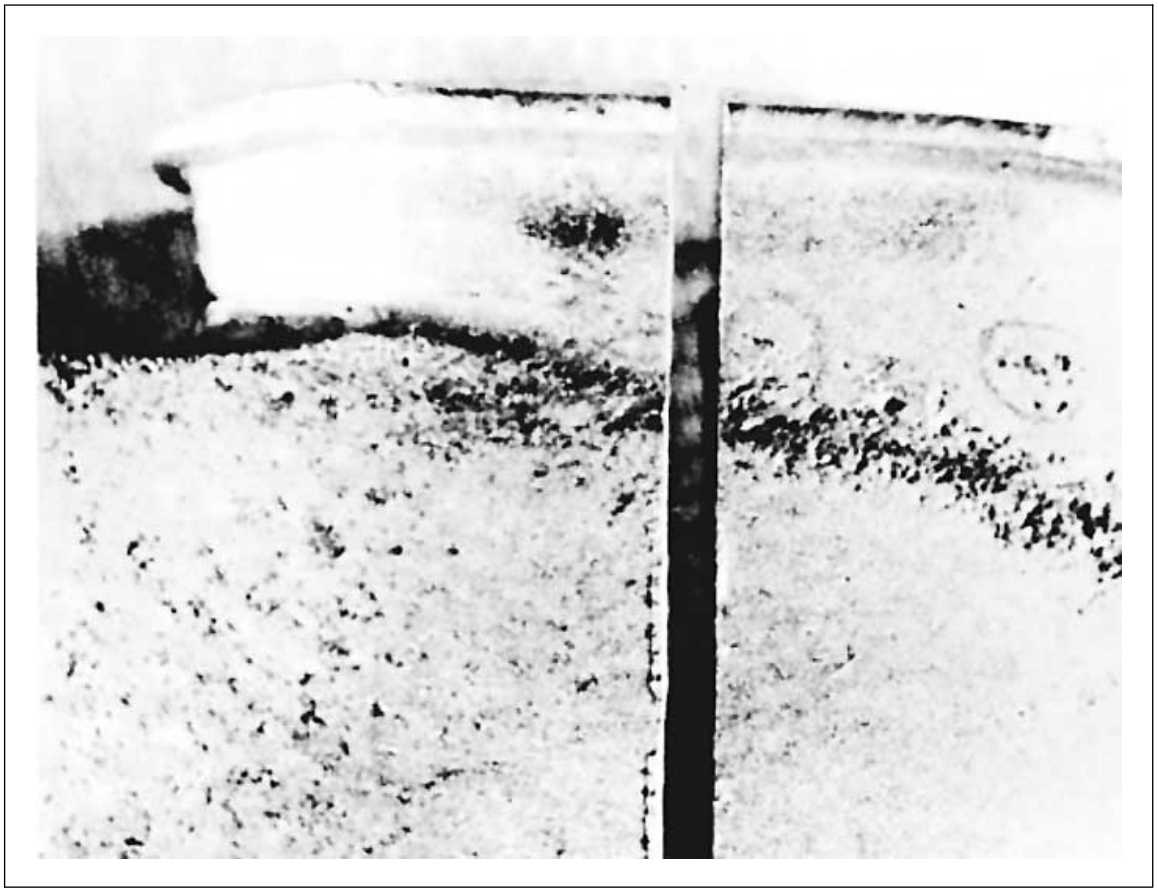

Fig. 5. Pitting in tenon of LP turbine blade.

For corrosion fatigue to progress into a blade failure requires both blade resonance and an advanced degree of corrosion. Therefore, unless the existence of blade resonance was known (not determinable by NDE), merely noting the existence of corrosion alone would not justify the expense of replacing the blades.

In addition to sodium chloride, hydrogen sulfide also is considered a possible corrodent of the turbine blades. If sodium sulfite is used as an oxygen scavenger in the boiler water, care must be taken to control sulfides in the steam by means of neutralizing additives such as phosphates. All volatile treatments (AVT) or oxygenated water treatment (OWT) for once through boilers are feasible alternatives.

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In summary, the most cost-effective approach to avoiding corrosion fatigue failures of LP turbine blades is by control of chloride ions or sulfides in the steam, and by monitoring steam purity on a continuous basis. Inspection is at best a supplemental approach because it would not be cost-effective to conduct inspections with sufficient frequency.

3.2.6 Bearings and Lubrication

Turbine lubricating oil systems are designed to supply clean, filtered and reconditioned oil to journal and thrust bearings at proper pressures and temperatures.

These systems need to be thoroughly cleaned of all debris and free of corrosion prior to start up (e.g., hot oil flushing, ultrasonics). Clean all filters and screens when necessary during normal operation. Take and analyze oil samples periodically to detect the presence of excess moisture, metallic particles or dirt. The presence of excessive contamination in the form of particulates could scratch or score bearing surfaces. When scratching and scoring become widespread on bearing surfaces, the oil film may break down, permitting contact between the stationary and rotating members, wiping the bearing.

Incipient wiping of a bearing may sometimes be detected by monitoring the temperature of the oil on the discharge side of the bearing. Thermometers and/or thermo couples are installed in bearing cap wells or in drain lines for this purpose. The discharge temperature is based on the oil cooler out temperature plus a bearing temperature rise. Use manufacturer's recommendations for these various values. An abrupt increase in bearing temperature may indicate that wiping is taking place.

3.2.7 Shaft and Coupling Cracking

Steam turbines have been subject to cracking of the shafts at couplings (see Fig. 6) and at the fillets between the shafts and the spindles.

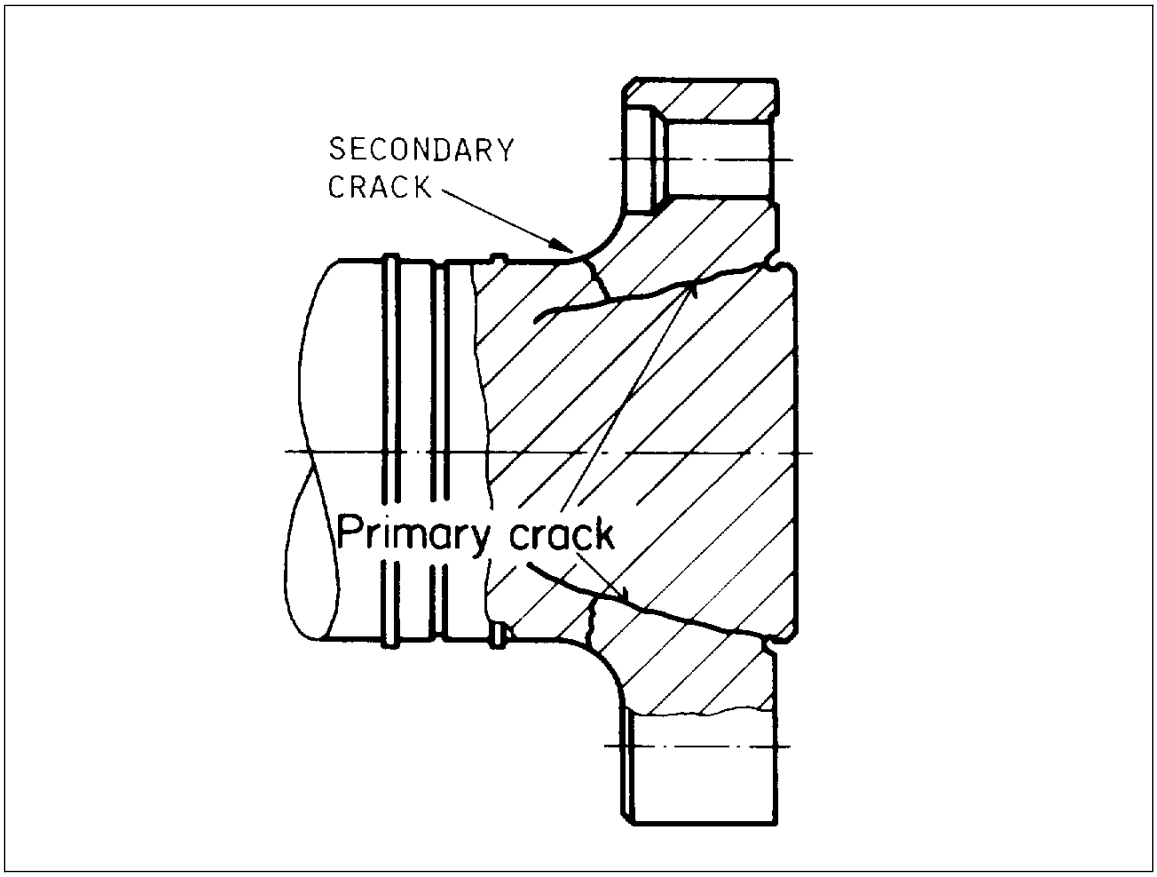

Fig. 6. Cracking in coupling of a utility turbine-generator.

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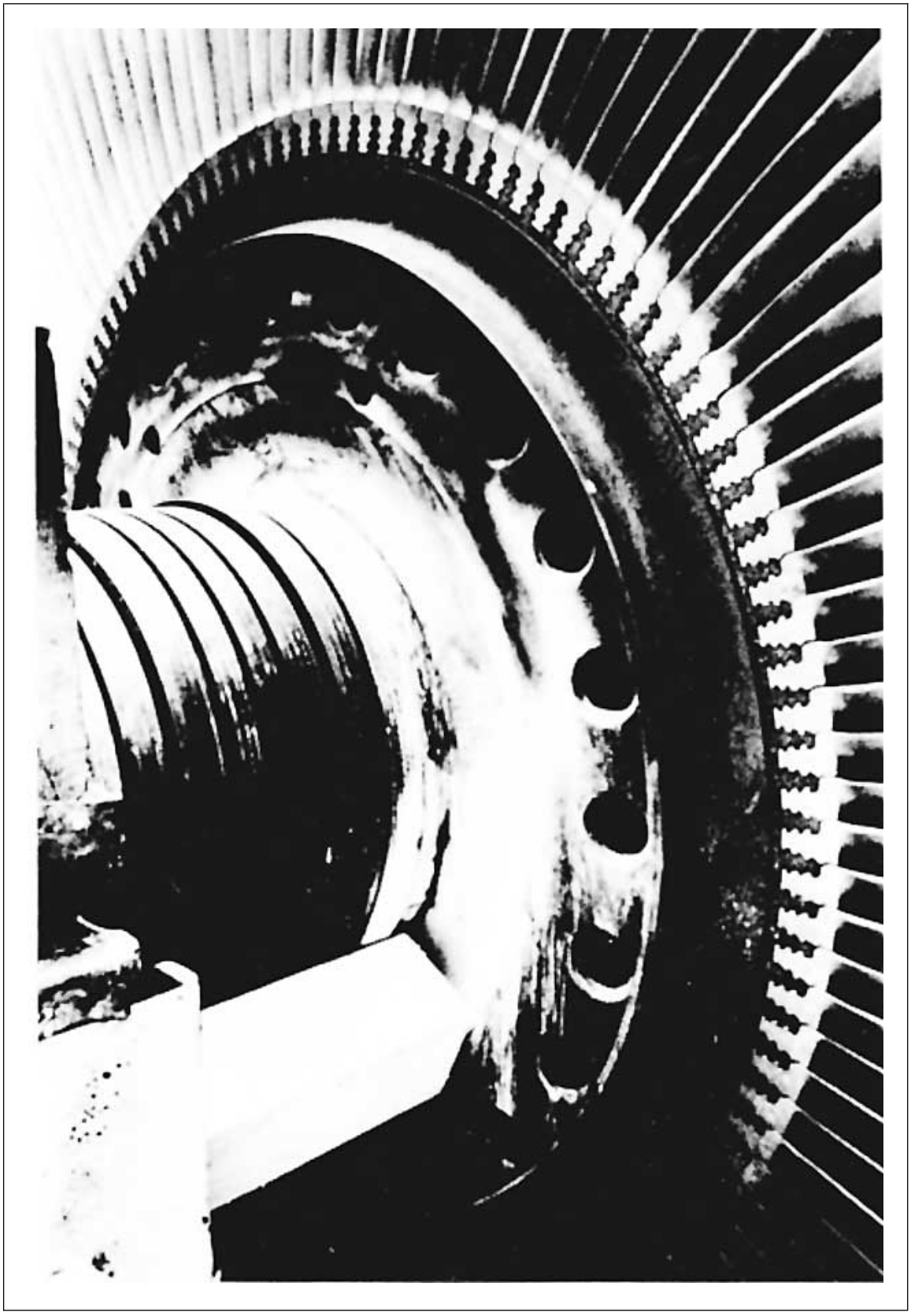

Fig. 7. Cracking in shaft fillet at spindle. (Bently-Nevada)

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The cracking in Figure 6 is a well documented case of fatigue cracking originating in corrosion pitting in the recess groove (stress raiser)(Bibliography, Ref. 2). It is believed that a sulfur-containing corrodent was introduced into the cavity between the coupling flanges during erection. The alternating stresses that produced the fatigue are believed to be caused primarily by misalignment. Reference 2 shows that the level of stress produced by the maximum permissible erection tolerances could result in a very slow crack progression in a corrosive environment, indicating that corrosion was the primary cause of the cracking in this case. However, it is also possible that misalignment had been aggravated during operation since erection.

Water cutting has been a factor in other cases of cracking in the fillet radii of coupling flanges (at the secondary-crack location in Figure 6). Condensing steam leaking through the adjacent steam gland was a chronic problem in the case of one manufacturer's three-bearing LP-turbine/generator sets. Some water-cutting is probably inevitable, which reduces the fatigue strength of the shaft in the fillet area, making it more susceptible to cracking if misalignment exists. The approach of the manufacturer has been to place very heavy emphasis on periodic alignment checks to avoid this hazard.

Utility steam turbines also have experienced cracking as shown in Figure 7. This is not usually associated with corrosion or water-cutting. There have been some material defects (porosity) in the fillet areas of some of these large forgings, which reduce the fatigue strength of the material to the point where bearing offset misalignment can produce cracking.

Figure 8 shows the principle of fatigue in a shaft due to offset misalignment. Offset misalignment occurs if one or more bearing(s) in a train of three or more bearings is (or are) not perfectly in line with the others. The alternating tension and compression in the shaft at a location of stress concentration such as a coupling flange fillet radius will, after 10^6 or more revolutions, result in the initiation of a crack. Additional cracks may develop and progress, linking up with each other and leading eventually to complete fracture of the shaft or coupling.

It is essential to realize that there is no warning of cracking due to offset misalignment (no bearing vibration is produced). In the figure, bearing pedestal No. 2 has settled, pulling the shaft downward. A vibration pick-up placed on this bearing cap will not see an alternating load since the load on the bearing is always in the upward direction. Therefore, there will be no vibration response. The only way to prevent cracking of this type is to check the hot running alignment periodically at intervals based on historical data, and to correct alignment by shimming bearings as necessary.

3.2.8 Stress Corrosion

An example of the second type of cracking is shown in Figure 7.

Stress corrosion cracking in steam turbines occurs when a corrodent precipitates out of the steam and attacks disks or spindles under high stress. In the large fossil turbines it occurs almost universally in the LP shells where moisture is likely to form. The usual corrodent in a moist environment is a caustic (sodium hydroxide) which collects in blade attachment slots and attacks the material of the disk or spindles. These are areas of high steady stress, where cracks may form and progress to complete rupture.

In stress corrosion incidents of this type, experience has shown that relatively short intervals between inspections would be required to detect cracking at an early stage. All incidents of record have been associated with inadequate steam purity, sometimes with specific discovery of caustic carryover. It is believed that control of sodium in steam is the most reliable and cost-effective method of avoiding losses caused by stress corrosion.

Craze-cracking on disk faces has been experienced in some LP turbine rotors. This type of cracking has not been associated with any specific corrosive medium. The manufacturer has established inspection intervals for such rotors to monitor the progression (if any) of this cracking.

3.2.9 Embrittlement of Spindles

Impurity-induced embrittlement has resulted in fractures and cracking of turbine rotor spindles. This type of failure is not preventable by controlling steam purity or by performance monitoring. It is associated primarily with rotors manufactured prior to 1960, when vacuum-melting of forging ingots was not common practice.

There is no direct way of testing nondestructively for embrittlement. Ultrasonic examination of the bore (boresonic testing) can determine if cracking exists or is developing. These test results must be evaluated with great care to avoid unnecessary retirement of a rotor. ***To detect possible crack development and***

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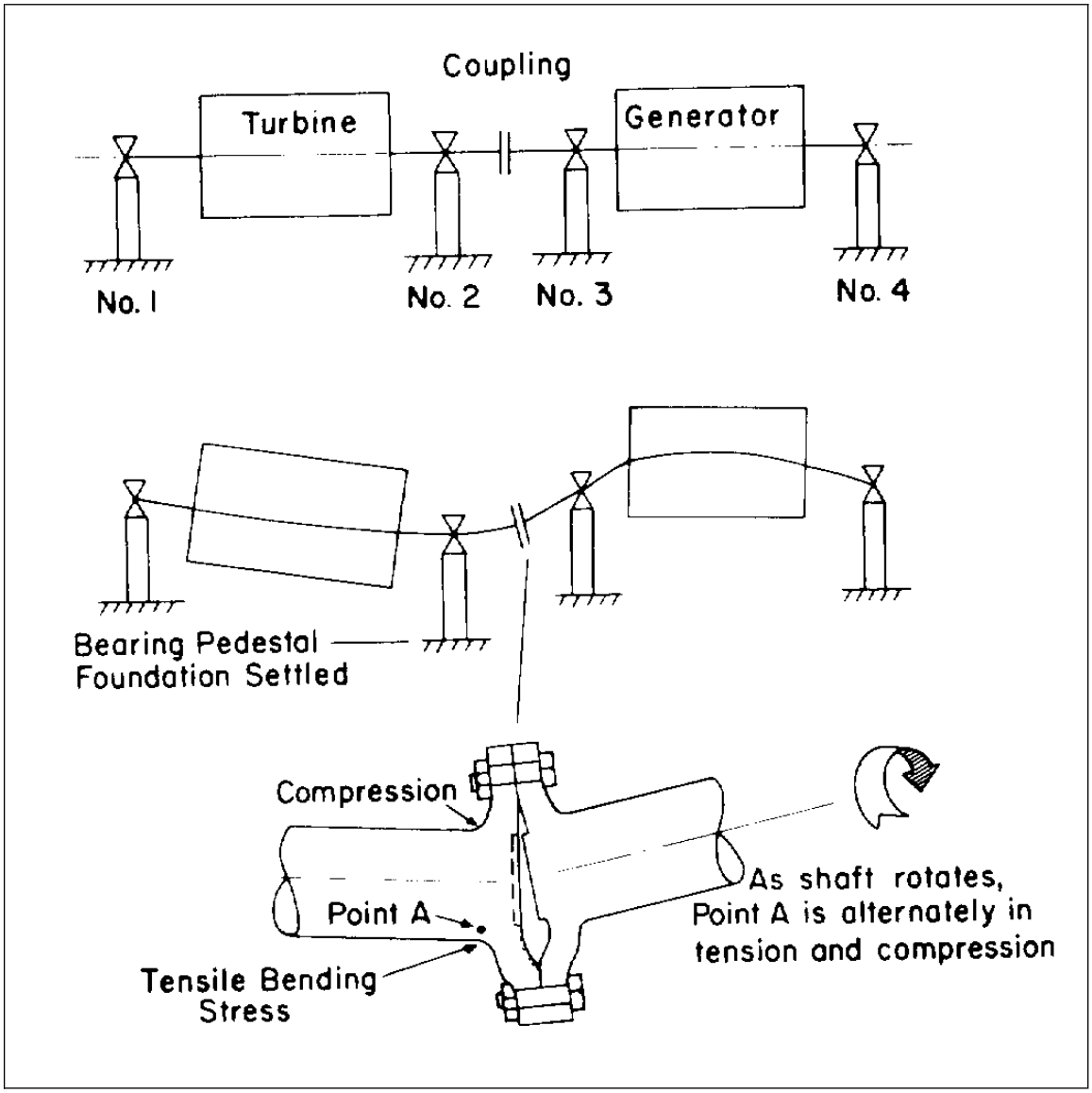

Fig. 8. Bending stress due to offset misalignment.

progression, conduct dismantle and boresonic inspections at intervals not exceeding five years on all rotors manufactured prior to 1960 and an OEM recommended intervals for bored rotors manufactured subsequent to that time. (Reference Section 2.2.9, bullet on NDE of disks and rotors).

3.2.10 Thrust Bearing Wiping

Thrust bearing wiping, other than that due to lubrication system failures and water induction, is caused by phenomena that transfer pressure drop from stationary diaphragms to the rotor. Such a transfer may result from accumulation of deposits on rotating blades or stationary vanes, severe distortion of stationary vanes, and/or deterioration of interstage labyrinth seals or packings. These phenomena usually lead to measurable drops in performance. In the case of the first two, both efficiency and flow capacity of the turbine are affected, while packing deterioration reduces efficiency only, with no effect on steam flow capacity. Thus, performance monitoring is an effective method of anticipating thrust bearing failures from internal causes. Depending on the thrust-balancing arrangements, high exhaust (condenser) pressure also can lead to thrust bearing wiping.

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Protection against thrust bearing wiping can be provided by thrust-wear detecting devices or temperature sensing devices embedded in the babbitt of the thrust pads. Several methods are described below.

3.2.10.1 Proximity Probes

Proximity probes are noncontacting. They are mounted so that they sense the proximity of the thrust-runner on the shaft (Fig. 9). As the shaft moves axially the gap changes; the field current changes correspondingly. These changes are detected by a monitor which can be set to alarm and/or trip the turbine.

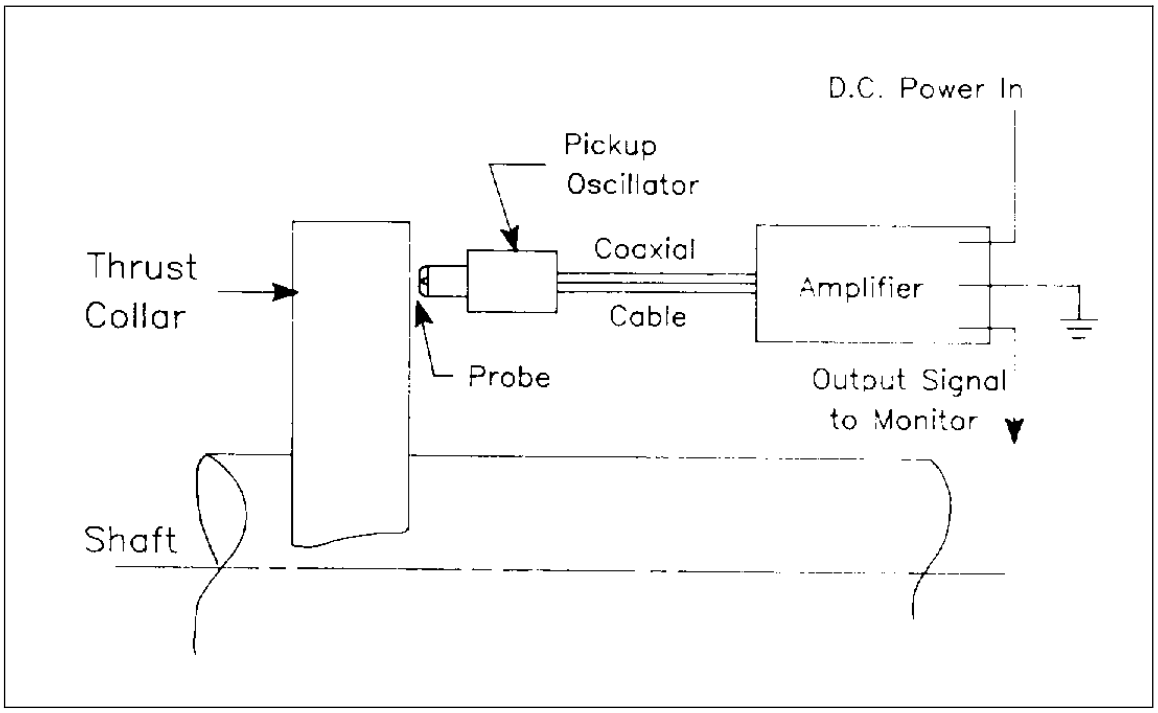

Fig. 9. Proximity probe installation.

3.2.10.2 Hydraulic Wear Detector

Figure 10 shows the installation of a hydraulic wear detector. As the shaft moves axially the nozzle on the follower piston moves in or out to maintain a constant gap. This is achieved by maintaining a pressure of one atmosphere in the nozzle and the internal balancing of the follower piston. The follower piston moves the pilot valve and the collars which maintain opening settings for a system of hydraulic ports depending on the position of the valve. These settings control the pressures in the lines to the pressure switches.

3.2.10.3 Embedded Temperature Detectors

The most sensitive approach to thrust wear detection involves measuring the actual temperature of the babbitt surface. Experiments have shown that, as the runner approaches contact with the thrust bearing, the temperature of the babbitt increases dramatically. Temperature detectors, thermocouples or resistance-temperature-detectors (RTD's), may be embedded in the babbitt as shown in Figure 11, set to actuate an alarm at a specified temperature, and a trip upon a temperature that implies imminent danger of the babbitt melting. The value of this system is that it does not make any difference what causes the rise in temperature; the bearing is protected against all eventualities that would cause severe wiping of the babbitt. Slight wiping may occur, but that is considered to be a small price to pay to protect against severe axial movement of the rotor into the stator.

Suitable alarm and trip settings for this protective scheme are: alarm at 210-220°F (100-105°C), and trip at 240-250°F (115-120°C).

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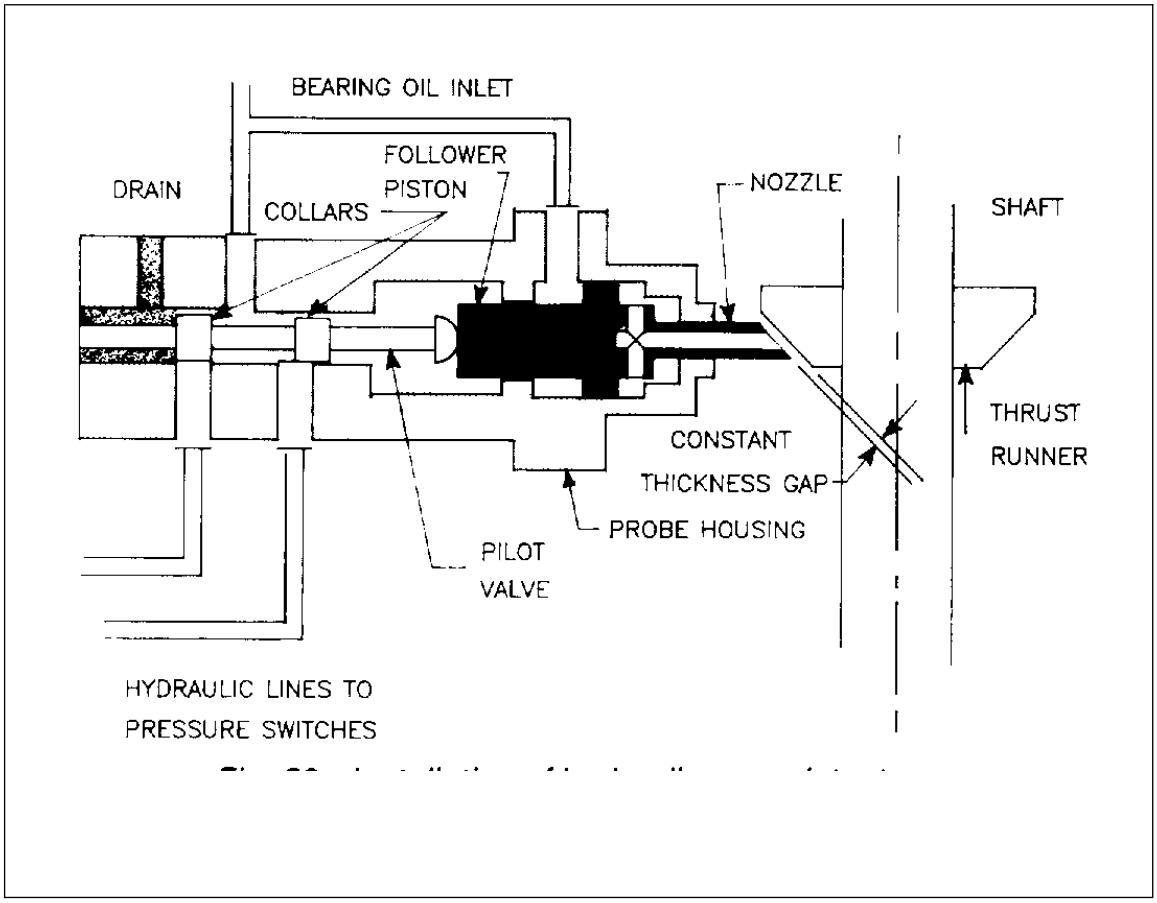

Fig. 10. Installation of hydraulic wear detector.

3.2.11 Low-Cycle Fatigue

In low-cycle fatigue, each cycle of operation produces plastic damage in turbine rotors until a crack is initiated. Such a crack may then progress under further cycles until the rotor is weakened such that it cannot withstand further centrifugal loading.

Turbine casings also may experience low-cycle (or thermal) fatigue. In these cases, since the strains producing the cracks are almost entirely the result of local thermal gradients, the strains disappear after a crack has progressed to some length, arresting the progression. Such cracks are usually detected as a result of steam leakage. They are readily repairable by welding or cold mechanical methods, and the casing will continue to function satisfactorily.

In rotors, low-cycle fatigue is much more serious because of the constant centrifugal forces present, both for contributing to the growth of a crack and for causing catastrophic rotor burst when a crack has progressed to a critical size. This was of particular concern in the electric power generation industry because of the increased use of turbine-generators in cycling or two-shift operation. Older turbines in base-load service were expected to put on a total of 700 starts over their desired 30-year life spans, and, of these, only 100 would be cold starts.

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Fig. 11. Temperature detectors embedded in thrust bearings.

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One industry estimate was that a two-shift turbine might experience over 8,000 starts in the same period. Startup cycles were carefully controlled to minimize thermal strain. However, some plastic strain does develop and each cycle of operation represents a measure of life used up in the HP rotor or IP rotor. The amount depends upon the details of a given turbine and on the particular load control system, but the following estimates may be used as a guide:

1. If the turbine is being started under control of a stress monitor:

Cold start:	0.05 life used.
Warm start (weekend shutdown):	0.05 life used.
Warm start (two-shift operation):	0.03 life used.
Hot start (following unit trip):	0.01 life used.

2. Older turbines without automatic stress monitor control, but with manual acceleration and loading ramp control based on estimated thermal gradients:

Cold start:	0.15 life used.
Warm start (weekend shutdown):	0.10 life used.
Warm start (overnight operation):	0.10 life used.
Hot start (following unit trip):	0.05 life used.

Note: Wherever available ***always*** use manufacturer's estimates.

The strategy for prevention of low-cycle fatigue fracture is based on rotor inspection after the estimated life used figures show that 80 to 100 percent of HP- or IP-rotor life has been used up. Therefore, inspection should include nondestructive examination of the turbine rotor surface, including the complete axial length of the blade retaining slots. The rotor bore should be inspected by ultrasonic techniques (boresonic inspection). Any patterns of indications (which might be cracks) should be machined out by bottle boring until it is clear that a crack either did not exist originally or no longer exists. Limits of radial material that may be removed without weakening the spindle excessively can be obtained from the manufacturer.

Subsequent inspections should be at a greater frequency. Even though no crack is found during the 100 percent cyclic life inspection, it is safe to assume that initiation may be imminent. Once a crack does initiate, each further cycle of operation causes a finite amount of growth until the critical crack length is reached.

It is not possible to provide guidelines for the number of cycles that may be applied after crack initiation before the critical crack length is achieved. Too much depends on the rotor configuration, the rotor alloy and on the thermal gradients that exist during the operational cycles. A fracture mechanics analysis should be undertaken, preferably by the turbine manufacturer but, if necessary, by a qualified consultant, to provide a maximum number of cycles of operation before subsequent inspection.

3.2.12 Summary – Failure Modes and Effects

Table 10 summarizes common failure modes experienced with steam turbines, many of which are described in the foregoing. Trained personnel using specialized tools and machines are required to properly diagnose many of the failure modes.

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Table 10. Failure Modes and Effects Summary – Steam Turbines

Failure Mode	*Component Affected*	*Symptoms/Results*	*Detection*	*Control & Prevention*
Brittle Fracture	LP and IP-LP rotors	Major damage to rotating and stationary components	NDE inspection for cracks, including porosity. Initial and periodic.	Controlled start-up, limited cycling. Match steam and metal temperatures. Pre-warm to FATT. QC during manufacture.
Stress Corrosion Cracking (SCC)	Rotor disk rims and steeples. (Particularly aging buckets and rims in LP turbine wet- dry (Wilson) region.) Susceptible to steam contamination.	Cracks/destruction of passivation layer high stress areas. SCC transitions into low and high frequency fatigue.	Surface degradation. NDE (UT) (difficult to detect) bore side examination possible. MT on exposed areas.	Control of water chemistry including oxygen. Limit cycling. Mat'l properties (low yield steels, strict control of manganese, vanadium, nickel and sulfur). In advanced stages, remaining life determined by finite element analysis, separation mechanism studies and probability failure analysis.
Stress Corrosion Cracking (SCC) High Cycle Fatigue	IP-LP and LP rotors and shrunk on disks	Axial cracking, cracking in key way and stress relieving slots	High Vibration.	Correct disk to rotor fit for design temperatures and stresses. Water quality, steam quality control. Avoid large diameter shrunk-on disks.
Creep Fatigue	HP and IP Rotors particularly those manufactured in the 50's and those reaching design life limits.	Poor quality near bore material.	NDE Ultrasonic through the central bore for welded rotors from outlying surfaces. Forged rotors with central bore supplemental magnetic particle and measure bore diameter over time. White light borescope. Boat sample material analysis.	Proper selection of material and heat treatment. Trend of periodic inspection results for remaining life analysis. Repair Grind out damaged material and repair with weldment. -or- Remove damaged portion and replace with a spool piece.
Thermal Low Cycle Fatigue	HP and IP Rotors	Surface Cracks Metal Softening	Destructive metal hardness measurement (Charpy) Thermal Stress Monitoring (TSM). Scanning Electron Microscope striation marks at high magnification.	Grind out damaged mat'l and repair by welding. FEA for remaining life assessment and Fracture Mechanics Analysis. Minimize cyclic operation. Install TSM to monitor stresses.
Torsional Stress-Rotors	Turbine rotor inspection, esp. at generator end and inspection of coupling bolts.	Rotor or coupling bolt failure. High vibration	Rotational vibration monitoring (RVM) device. Monitor electrical system frequency. Generator Torsional Monitor. Variations in angular frequency of the turbine generator system.	Remove unit from the line if frequency is out of range. No out-of-phase synchronization. Verify machine double frequency resonance for units in service prior to 1975. Proper machine alignment.

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Failure Mode	*Component Affected*	*Symptoms/Results*	*Detection*	*Control & Prevention*
Rotor Unbalance	Rotor and its component parts	High Vibration	Vibration monitoring and diagnosis analysis along the entire shaft	Add balance weights and/or repair damaged parts. Check balance after overhauls.
Rotor Misalignment	Rotating System	High frequency vibration Radial, angular or miscellaneous displacement between coupled rotating sections	Vibration monitoring and analysis of the vibration in the horizontal, vertical and transverse direction, frequency spectrum, phase angle, validation along shafts.	Correct realignment. Follow assembly procedures. Proper analysis, design, location of bearings for all operating conditions, esp. thermal expansion.
Rotor Shaft Thermal Expansion/ Contraction	Rotor Assemblies	Axial rubbing between stationary and rotating parts. Possible variation in thrust bearing vibration	Differential expansion monitoring Thermal Stress Monitoring (TSM) system.	Match steam and metal temperatures.
Rotor bowing	Rotor Assemblies	High frequency vibration uneven circular temperature of the rotor assembly. Excessive clearance between adjacent disks for units with shrunk on disks.	Vibration monitoring and analysis. Casing upper and lower differential temperature.	Provide water induction protection. Use turning gear. Isolate rotor from heating/cooling sources during shut down. Maintain steam seals until vacuum has been broken.
Rotor Low Frequency Vibration	Rotor Assemblies	Natural frequency is close to ½ rotating frequency. Bearing oil whip. Vibration oscillations.	Vibration monitor and lubricating oil temperature trending. Rotor design too stiff.	Monitor vibration and oil temperature, maintain within limits. Perhaps slightly increase oil temperature to reduce viscosity. Check for misalignment on multi element machines.
Water Induction	Stationary and Rotating Components	High vibration high differential temperatures upper and lower casing. Thrust bearing failure. Warping, distortion failure of rotating and stationary parts.	Rapid changes in system conditions include pressure, temperature, level, and feedwater and attemperator flow rates, firing rate. Thrust bearing vibration activity. Feedwater heater high and extremely high level. Audible signs of thermal shock and flashing.	Prevent water ingestion from all sources. Verify OEM provisions for prevention are implemented. Isolate the potential source(s). Continue to run the turbine if possible at load; reduce to turning gear speed as soon as possible and leave on gear.
Solid Particle Erosion	HP and IP blades and vanes	Wastage of trailing edges of blades and vanes, esp. high temperature/high pressure velocity sections of HP and IP turbine.	Efficiency Loss Borescope and fiber optic examination.	Repair/replacement of damaged parts. NDE. Chemically clean boiler Protective coating (limited and can cause other problems).

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Failure Mode	*Component Affected*	*Symptoms/Results*	*Detection*	*Control & Prevention*
Solid Particle Impact	Mechanical damage to stationary and rotating parts	Vibration Performance degradation.	Vibration monitor. Rapid change in operational data (performance).	Component repair or replacement. Proper assembly and securing of internal parts such as radial seal strips. Inspections of parts for mechanical integrity. Stop valve strainers following steam line and boiler maintenance.
Fretting Fatigue Initiated by Partial Arc Admission	HP Turbine Blades	Blade cracking that initiates at the blade root away from the root fillet area. Can also occur in the blade root area because of differential movement between the blade and disk.	NDE during outages.	Repair, replacement of damaged blades. Blade redesign if significant operation is anticipated. Switch to sliding pressure. Combined steam admission governing.
Low and High Cycle Thermal Fatigue	Inlet Sleeve and Seal Assembly	Sleeve cracking and seal assembly damage. Increased IP turbine exhaust temperature. Decrease in first stage pressure and temperature.	Magnetic particle testing. Design review. Open casing inspection. Ultrasonic examination on closed, cooled casing.	Restrict fast start-ups and rapid load changes. Sleeve repair (some repairs and blending may be possible without opening the casing). Replacement. Sleeve re-design.
Steam Contamination	Turbine blades esp. LP section in transition zones (dry to wet steam)	Performance degradation.	On-line quality monitoring of feedwater and steam. Deposit sampling from blades.	Follow OEM recommendation for water/steam chemistry. Common contaminants: Silicon Chlorine Sodium Organic acids, and carbon dioxide. Minimize attemperator use. Condensate polishing. Prevent condenser tube leaks. Upgrade blade material.
Moisture Erosion	LP turbine blades and vanes.	Trailing edge erosion	Borescope examination during shut down. Eddy Current test (EPRI)	Additives to steam to reduce droplet size. Add erosion strips. Change blade material. Diminish low load operation when the steam quality is the wettest.
Vibration Stress Cracking	LP Turbine Blades	Minute cracking at leading and trailing edges. Vibration induced by low volumetric steam flow.	Ultrasonic testing (Requires rotor removal and immersion in a test tank in which it is rotated). Detect dynamic stressing of last stage blades.	Avoid operating loads that place rows of blades in a natural frequency band. Repair/ replace damaged blades. Set minimum steam flow limits. Blade redesign following CFD or FEA. Minimize operations with FW heaters out of service. Raise rated last stage enthalpy drop.

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Failure Mode	*Component Affected*	*Symptoms/Results*	*Detection*	*Control & Prevention*
Low cycle thermal fatigue, creep, erosion/ corrosion	Stationary components- casing, diaphragms, cross over piping	Growth and propagation of initial casing flaws, esp. in highly stressed areas (L.C. thermal fatigue). Plastic deformation/distortion (creep).	Visual, magnetic particle. Ultrasonics. Bent diaphragms esp. in HP	Repair (Grinding, welding). Use Thermal Stress Monitoring (TSM) system if installed. More resistant material. Maintain high steam quality. Thermal barrier coating.
Abrasion, Cavitation, Corrosion, Pitting, Overheating, Wiping, Fatigue.	Journal and Thrust Bearings.	Electrical pitting. Loss of babbitt, babbitt scouring.	Visual rise in oil temperature leaving the bearing. High vibration. Lube oil analysis.	Eliminate electrical fields (residual magnetism) esp. after welding or Eddy Current testing of the unit. Provide shaft-grounding brushes. Decrease oil temperature to re-establish a wedge. Monitor and trend properties on a regular basis.

4.0 REFERENCES

4.1 FM Global

For associated fire related information on turbine-generators refer to Data Sheet 7-101, *Fire Protection for Steam Turbines and Electric Generators.*

A partial list of documents related to steam turbines and driven equipment follows:

Data Sheet 5-12, *Electric AC Generators.*

Data Sheet 7-100/13-16, *Dynamic Compressors.*

Data Sheet 12-2, *Pressure Vessels.*

Data Sheet 12-17, *Watertube Boilers.*

Data Sheet 13-7, *Gears.*

Data Sheet 13-18, *Industrial Clutches and Clutch Couplings.*

Data Sheet 13-24, *Fans and Blowers.*

Data Sheet 17-4, *Monitoring and Diagnosis of Vibration in Rotating Machinery.*

APPENDIX A GLOSSARY OF TERMS

Isentropic: [Thermo] Having constant entropy; at constant entropy.

Partial-arc admission: Admission of steam into a steam turbine through only a part of the steam inlet nozzles.

Steam turbine: [Mech. Eng.] A prime mover for the conversion of heat energy of steam into work on a rotating shaft, utilizing fluid acceleration principles in jet and vane machinery.

APPENDIX B DOCUMENT REVISION HISTORY

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APPENDIX C BIBLIOGRAPHY

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